AMERICAN MACHINIST

A JOURNAL FOR MACHINISTS, ENGINEERS, FOUNDERS, BOILER MAKERS, PATTERN MAKERS AND BLACKSMITHS.

VOL. 15, NO. 37. WEEKLY. | NEW YORK, THURSDAY, SEPTEMBER 15, 1892. | $3.00 per Annum. SINGLE COPIES 6 CENTS

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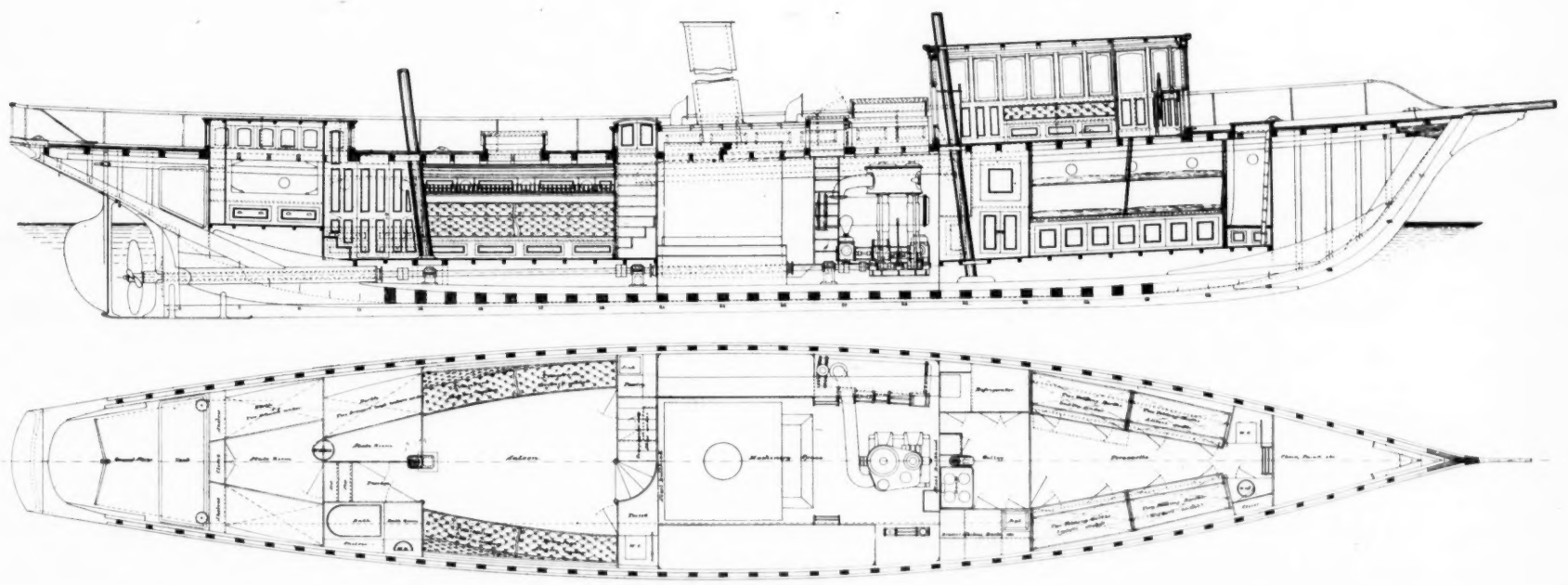
Steam Yacht "Theresa."

With this we give a longitudinal section and plan of the steam yacht "Theresa" recently designed by Henry J. Gielow, Engineer and Naval Architect, 29 Broadway, and built for Gus Sidenberg, Esq. She is a very fast boat of her class, having covered 15.8 miles during a continuous run of one hour. The length of boat over all is 95 feet; length on water line, 78 feet; extreme breadth of beam, 15 feet; draught, 6 feet.

The engine is of the inverted cylinder surface-condensing compound type, with two cylinders and cranks placed at an angle of 90 degrees. The cylinders stand fore and aft of each other, the high-pressure cylinder being the forward one; its center is 4 inches forward of the center of frame 29, and the center of shaft is 2 feet 9 inches below the designed water line on the same frame.



STEAM YACHT "THERESA."

The high pressure cylinder is 9½ inches diameter; low-pressure, 19 inches diameter; and stroke, 14 inches. Steam ports of high pressure are 1½ inches wide and 7½ inches long; exhaust port, 3⅝x7½ inches. The width of steam and exhaust ports in the low-pressure cylinder are the same as in the high-pressure cylinder, and the length is 14 inches. Thickness of metal in H. P. cylinder is 1/16 inch and ⅝ inch in L. P. cylinder. On the port side the cylinders are supported by cast-iron frames, and on the other side by two wrought iron columns 2½ inches diameter at the base and tapering to 2¼ inches at top. For the distribution of steam in the H. P. cylinder a piston valve 5½ inches diameter is used, and an ordinary D valve performs the same office in the L. P. cylinder.

The intermediate receiver consists of a copper pipe 6 inches inside diameter and of sufficient thickness to stand a steam pressure of 150 pounds.

The valves are worked by a "Stephenson" link motion. The links are of the open pattern, with a throw of 12 inches; they are made of machinery steel 1 inch deep and 2 inches face.

The width of face of eccentrics is 2 inches; eccentricity, 4 inches. The eccentric rods are made of machinery steel of rectangular section, ⅝x1⅝ inches in neck near the links, and ⅝x2 inches near eccentrics; the link ends are forked, and the eccentric ends are of the flat bar form, ⅝x2½ inches, fitted in corresponding slots in the eccentric straps; the bolt holes are oblong, so as to admit of adjustment for length. The valve stems are of mild steel 1⅛ inch diameter for the L. P. cylinder and ⅞ inch diameter for the H. P. cylinder. The lower portions of valve stems are made rectangular and work in composition steady rests. The cast-iron pistons are fitted with followers secured with steel bolts screwed into brass nuts. The packing rings are of cast-iron in two thicknesses and set out with steel springs. The piston rods are of mild steel 1⅝ inches diameter.

The crosshead is of cast-steel; journals are 2½ inches diameter and 3 inches long. The crosshead is guided by slipper slides of hard composition, babbitted, working in cast iron guides. The bearing surface of the back of slipper for each cylinder is 52 square inches.

Connecting rods are made of hammered iron with solid eye at crosshead end and strap and key at crank-pin end. They are 36 inches long and furnished with composition boxes at each end, lined with magnolia metal and fitted with tapered keys, set-screws, etc. They are 1¾ inches diameter at crosshead end and 2½ inches diameter at crank-pin end; the sides are formed by two parallel planes 1⅝ inches apart.

The crank-shaft pillow-blocks are fitted with composition boxes lined with magnolia metal. The forward journal is 5⅝ inches long; the after and intermediate journals are each 9 inches long, and all are 4 inches diameter. The caps are recessed for the boxes, and held in position by ten steel bolts 1¼ inches diameter; two in forward bearing, four in middle and four in after bearing. The nuts are recessed into caps and held with set-screws; bolts are spaced 7⅝ inches between centers.

The crank-shafts are of wrought-iron forged in one piece and cranks set at an angle of 90 degrees. Crank arms are 2⅝ inches thick and 5⅝ inches wide. Crank-pin journals are 4⅝ inches long and 4 inches diameter. Cast iron counterbalance weights are secured to crank arms. The line shaft is made of compressed steel 3¾ inches diameter and made in three lengths, covered with composition at stuffing box and stern bearing. The length of outboard piece is 17 feet 6 inches; intermediate, 13 feet 6 inches, and forward piece 13 feet 9 inches.

The couplings consist of cast-iron disks, and fastened together by six turned steel bolts 1 inch diameter. The stern bearing is 15 inches long, lined with lignumvitæ. The hole through stuffing box log is 5½ inches diameter, lined with a lead sleeve ¼ inch thick.

The propeller wheel is of cast-iron, four-bladed, 4 feet 8 inches diameter and 7 feet 6 inches pitch; it is fitted on shaft with fore and aft key, and a cap outside with keeper. Steam is furnished by a Roberts water tube boiler, having a grate surface of 33 square feet, and 1,090 square feet of heating surface.

The hull is built of wood. The keel is of white oak, sided 6 inches and molded 8 inches on frame 38 and 10 inches at stern; it has a scarf 51 inches long with 2 inch nibs, and fastened with four yellow metal bolts ⅞ inch diameter, riveted over composition rings, and two composition spikes in each nib.

The stem is of white oak, natural crook, sided 6 inches, and extends 4 inches outside of rabbet. It is joined to the continuation of keel by a scarf 48 inches long, 3-inch nibs, fastened with four yellow metal bolts ⅞ inch diameter, riveted over composition rings and two composition spikes in each nib. The stem is bearded from rabbet to 1 inch in thickness at forward edge, and fitted with a brass cut-water.

The forward deadwood is of white oak, sided 6 inches; the fastenings are of ⅞ inch yellow metal, about 15 inches apart; the greater number of the bolts pass through and clinch over composition rings.

Stern post is of white oak sided 6 inches at heel and 10 inches at cross-seam and center of shaft, molded 12 inches and secured to keel by tenon and locust treenails.

The after deadwood is of yellow pine, sided 6 inches; the stuffing box log is of yellow pine 10x12 inches, made in two depths or layers with flannel between them; the lower piece is fastened with toe bolts, 15 inches apart, on each side of shaft well, and the upper piece by clinch bolts between the lower bolts to the lower piece; all bolts are of ⅞ inch galvanized iron through the whole deadwood, and those passing through the stern post are of yellow metal.

The rudder post is of 2½ inch round iron, runs up to the horn beam and fastened there with four ⅞-inch screw bolts. The base is forged to fit a 6 inch iron shoe, extending 4 feet 6 inches forward of after end of keel and bolted to same with two ⅞-inch screw bolts. The iron is fastened to keel with four ⅞-inch galvanized iron bolts; heads are countersunk so as to be flush with bottom; jaws are forged on and extend 2 feet up on each side of stern post and securely bolted there.

The rudder stock is of 3 inch round iron, outside rim 2 inches square iron drawn down to 1x2 inches at after edge, filled with white pine and pitch and plated with 5/16 inch iron.

The frames are of hackmatack, placed 22½ inches from center to center, double-framed; each is sided 4 inches in floors and first futtocks and 3 inches balance of way up, and molded 6 inches at keel and 3 inches at plank sheer. The timbers composing the frames overlap each other not less than 2 feet, with two ½ inch bolts to each butt. Floors are of natural crooks and have alternate long and short arms. A bolt of ⅝-inch galvanized iron passes through each floor timber into keel, driven alternately 2 inches from each edge of keel. The heels of cant frames against the deadwood box in ½ inch, and each is secured with two 7-inch galvanized iron spikes. The space between the frames of the square body are filled in solid with white pine, the ends rising about 3 inches above bottom of keelson, and calked on outside and inside.

The keelson is of fat pitch pine 6 inches square; scarf is 7 feet in the clear of keel scarf, of equal dimensions and fastened with four ⅝-inch galvanized iron bolts clinched over galvanized iron rings. The fastening of the keelson is one ⅝ inch galvanized iron bolt passing through each frame into keel and one ⅝-inch yellow metal bolt passing through each frame and keel and clinched over composition rings The outside planking is of

yellow pine 1¼ inch thick before fitting and fastening. Sheer strake below the plank sheer is 8 inches wide. No plank exceeds 9 inches in width, and all plank above water line does not exceed 4½ inches in width, and are fastened with one 4-inch spike and one ½-inch through bolt; below water line, one 5-inch spike and one ½-inch through bolt, and with one additional spike to each frame in plank more than 6 inches. The bolts are clinched over rings on inside of ceiling. The butt bolts, two to each butt, except at ends of boat, where one is used, are ½ inch diameter and clinch over rings on the inside of ceiling. All fastenings from keel up to 6 inches above water line are of yellow metal, through bolts of copper, the remainder of galvanized iron. All seams are calked with one thread of cotton and one thread of best navy oakum white-leaded and puttied. Parallel to and 2 inches below plank sheer there is an oak wearing strake 3x4 inches amidships and tapering to 1x4 inches at ends.

Bilge strakes are of yellow pine three on each side at turn of bilge, 6 inches wide and 3 inches thick, nip into clamps on after ends, and bolted edgewise to each other in every space between the timbers by ½-inch iron blunt bolts; each strake is fastened to each frame by two 6-inch galvanized iron spikes besides the bolts passing through from outside. The ceiling is of yellow pine ⅞ inch thick and fastened with two 3-inch galvanized iron spikes to each frame.

The beams are of white pine molded 5 inches at center and 4 inches at ends, worked with a crown of 5 inches amidships. They are sided 4 inches. On frames 6, 11, 19, 27, 30 and 38 the beams are sided 8 inches, and intermediate beams are sided 3 inches.

The deck plank is of dry, well seasoned white pine 1¾ inches square, laid parallel to sides of boat. They are fastened with 4-inch iron deck spikes, driven near alternate edges one to each beam, white-leaded and plugged. Seams are calked with two thread of cotton white leaded and puttied. The lower deck is of white pine ⅞ inch thick and 4 inches wide. A mahogany rail extending all around the vessel is supported by ¾-inch galvanized pipe stanchions 24 inches long, spaced about 5 feet apart. The space between rail and deck is fitted with rope netting.

Masts are of white pine. Foremast from deck to hounds, 29 feet; diameter, 8 inches; mainmast from deck to hounds, 30 feet; diameter, 8 inches; fore topmast, 23 feet long; main topmast 23 feet long; fore gaff of spruce, 17 feet long; main gaff of spruce, 18 feet long; both have a diameter of 4 inches; white pine bowsprit, 6 inches diameter; fore and main sails to brail to masts; mastheads, 5 feet. Sails and awnings are of cotton canvas. One foresail of No. 8; one mail sail of No. 8; one fore stay sail of No. 8; one main stay sail of No. 7, and one jib of No. 8 cotton canvas.

The pilot house is 16 feet long with sides parallel to and 29 inches from sides of vessel; the front is circular.

The stateroom trunk is 9 feet 3 inches long, 7 feet 2 inches wide forward and 4 feet 10 inches aft, and 2 feet high. The sides have sashes fitted with ground glass lights.

Forecastle is 15 feet long, extending full width of vessel, fitted with four folding berths 6 feet long and 24 inches wide, with lockers under, along each side.

The saloon extends full width of vessel and 12 feet fore and aft with transoms 18 inches wide along each side with lockers underneath. The passage leading from saloon to owner's stateroom is 2 feet 6 inches wide.

The small stateroom on the port side of passage is fitted with one berth 30 inches wide and 6 feet long.

The owner's stateroom at after end of passage, and reached by two steps, extends full width of vessel and 7 feet fore and aft. Each side is fitted with a berth 30 inches by 6 feet, with two drawers under and a small closet and shelves at head. All wood-work is of mahogany and polished. A bath-room is located opposite the small stateroom, fitted with zinc bath tub, and plumbed for fresh and sea water. It is also connected with boiler for heating water, and has all necessary pipes, nickel plated valves, faucets, etc. The water is discharged into a receiving tank under floor, and connected directly with bilge ejector for discharging overboard. The whole inner arrangement has been designed for comfort and health, and is a credit to the designer.

Centrifugal Action in Ropes and Belts.

BY M. E.

The recent epidemic of fly-wheel failures has had the effect of redirecting attention to the stresses induced by centrifugal force in pulleys, wheels and revolving bodies generally, when rotating at high speeds, and as a consequence several more or less instructive investigations of the subject have been recorded during the past few months. In almost every instance, however, the main object of the inquiry has been the calculation of the stresses which are set up in wheel rims running at high velocities, and it appears to the writer that the consideration of the subject of centrifugal action might be extended with advantage, more particularly with regard to its influence upon the transmission of power by such flexible connections as ropes and belts. In one sense, the latter aspect of the question is of greater importance to the power user, for so long as a fly-wheel rim possesses sufficient strength to just resist the disruptive effects of centrifugal force, neither breakdown or loss of power occurs. But if from the rim of this same wheel power is transmitted by belts or ropes, it is more than likely that the high circumferential speed of the wheel may involve a considerable loss of efficiency in the power transmission, increasing the wear and tear of the belts or ropes employed, and entailing a very great risk of breakdown by the failure of these elements.

In the present contribution it is the writer's object to describe a graphic method of investigating the effects of centrifugal force upon belts and ropes at various speeds, and especially to emphasize the fact that all such flexible transmitters have each a limiting velocity at which, with a given intensity of stress, the power transmitted will reach a maximum amount. As a general rule, the belt manufacturer appears to tacitly ignore this point, for the tables of power transmitted by belts and ropes as given in most catalogues are based upon the assumption that, with a given intensity of stress, the power transmitted increases directly as the speed, other conditions remaining unaltered. When the velocity of transmission is comparatively low the assumption may be allowed, but at high speeds the effect of centrifugal force in flexible transmitters becomes very great, modifying to a very considerable extent the conditions which are commonly supposed to obtain. With the constant tendency towards the use of higher speeds, the full recognition of this fact becomes increasingly necessary, and in all high-speed systems of power transmission due regard must be paid to the effect of this disturbing influence, if satisfactory working is to be obtained. In any case, the manufacturers of belts and ropes might exhibit a little more intelligence in the compilation of their tables of power transmission. At present, a belt running at 6,000 feet per minute is considered capable of transmitting ten times the power that it would carry if running at 600 feet per minute. As will be presently shown, however, this actually implies that the belt is either capable of transmitting more power at the lower speed, or that it is overloaded at the higher rate. Certainly the conditions of maximum efficiency cannot obtain in both cases.

Fig. 2.

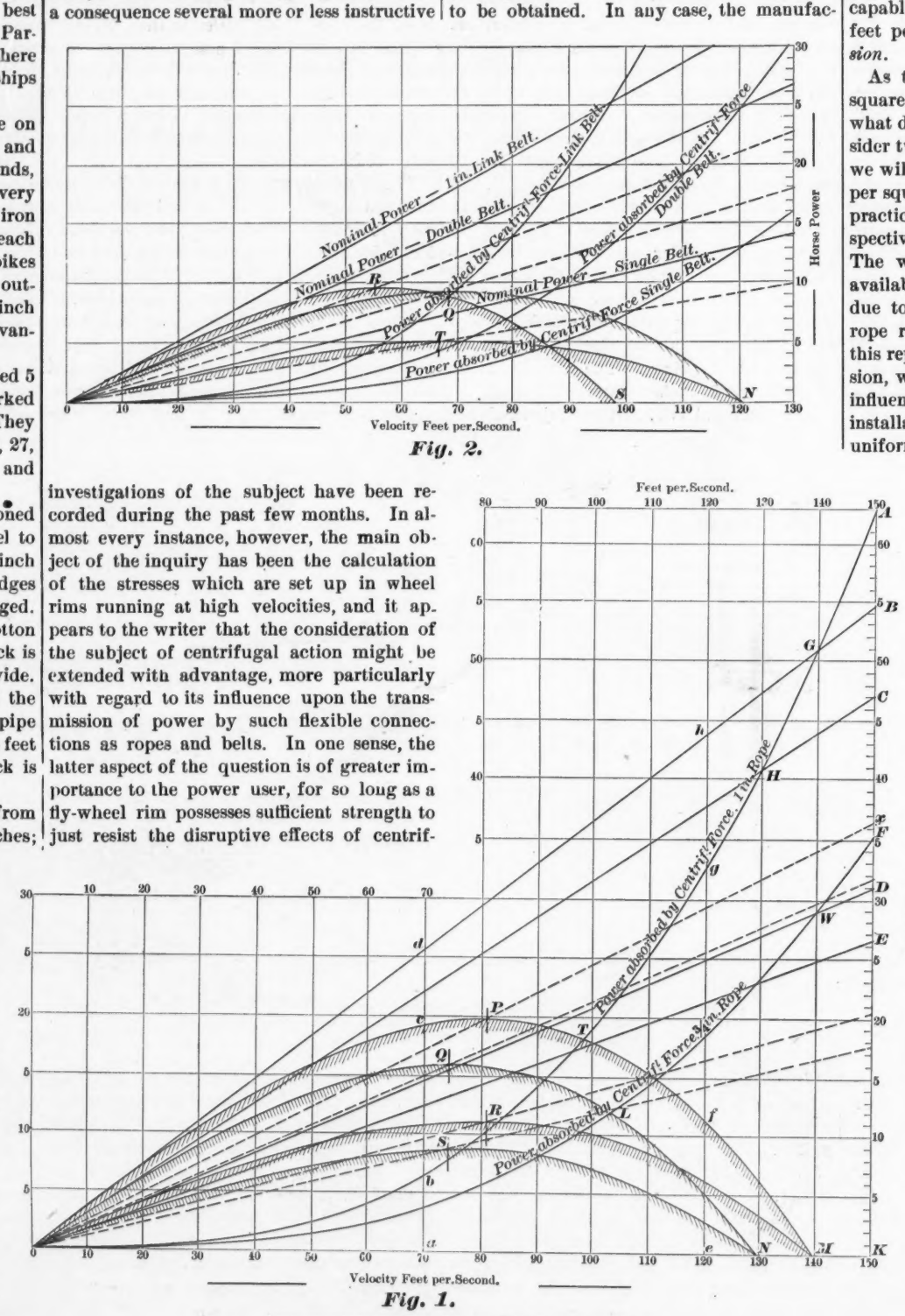

Feet per Second.
Power absorbed by Centrif. Force 1 in. Rope
Power absorbed by Centrif. Force ¾ in. Rope
Power absorbed by Centrif. Force ¾ in. Rope
Velocity Feet per Second.
Horse Power

Fig. 1.

CENTRIFUGAL ACTION IN ROPES AND BELTS.

The graphic diagram, Fig. 1, has been prepared to exhibit the effect of centrifugal action in the transmission of power by hemp ropes. For convenience, and to avoid confusion, only two sizes of ropes have been dealt with—three-quarter inch and one-inch diameter, but the method is it will be understood, equally applicable to all other sizes, and the diagram can be readily extended as desired. As will be seen, the horizontal base line is spaced off by a number of equidistant divisions, representing velocities of rope of from 0 to 150 feet per second, and from which ordinates are drawn, as shown. The ordinate *K A*, which corresponds to a velocity of 150 feet per second, is also spaced off, the divisions representing in this case the horse-power transmitted. This vertical horse power scale may be made of any convenient length, and subdivided as far as desired, but the main divisions, at least, should be emphasized by a series of horizontals drawn across the diagram, as indicated.

Dealing first with the one-inch rope, we proceed to determine the horse power it is capable of transmitting at a velocity of 150 feet per second, *disregarding centrifugal tension*.

As to the maximum allowable stress per square inch for hemp ropes, opinion is somewhat divided, so it will be advisable to consider two values, which, in the present case, we will take as 320 pounds and 275 pounds per square inch, corresponding to stresses of, practically, 250 pounds and 215 pounds respectively for a rope one inch in diameter. The whole of this tension is not, however, available for power transmission, as the stress due to the tension in the slack side of the rope requires to be deducted from it, and this represents a percentage of the total tension, which varies somewhat in value, being influenced by the care bestowed upon the installation generally, evenness of splicing, uniformity of length of the ropes, shape of groove in the wheels, etc. Probably about 15 per cent. represents the proportion of the total tension nullified in this way under the best conditions, but for the present purpose 20 per cent. will be taken, so that the driving tension in the foregoing instances is $\frac{250 \times 80}{100} = 200$ pounds, and $\frac{215 \times 80}{100} = 172$ lbs. respectively.

If, as assumed, the effect of centrifugal force is disregarded, the horse-power transmitted by each rope is:

$$H.\ P. = \frac{T \times v}{550};$$

T being the driving tension, *v* the velocity in feet per second, and *H. P.* the horse-power. In the first of the above cases, we have at a velocity of 150 feet per second,

$$H.\ P. = \frac{200 \times 150}{550} = 54.54.$$

And setting off 54.54 on the vertical scale *K A*, we obtain the point *B*, which is joined to *O* by the straight line *B O*. The lengths of the velocity ordinates intercepted between this line and the base *O K* represent on the horse-power scale the power transmitted at various speeds, on the assumption that the driving tension remains constant. Similarly for the lower value of stress assumed in the second case, we have, at 150 feet per second,

$$H.\ P. = \frac{172 \times 150}{550} = 46.9.$$

And setting off this value on the horse power scale, the point *C* is obtained, from whence *O C* is drawn, representing the power transmitted at the lower working tension, and at the speeds given. The lines *O D* and *O E* similarly represent the power transmitted by three-quarter-inch ropes, the same stresses per square inch as in the one-inch rope being assumed.

We have now to take into account the modifying influence of centrifugal force, and in order to exhibit directly its effect upon the efficiency of the transmission, the curve representing the power absorbed by centrifugal force at various speeds is superposed upon the diagram as shown. *O T A* is this curve for one-inch rope, and *O L F* that for three-quarter inch rope. The method of constructing these curves is very

simple, and may here be described, as a special curve is required for each diameter of rope. The stress due to centrifugal force in a belt or rope is:

$$S = \frac{w\,v^2}{32.2};$$

in which w = weight of a running foot of the material, v the velocity in feet per second, and S the stress per sectional area of rope. The *horse-power* absorbed by centrifugal force is, therefore, $H.\ P. = \frac{w\,v^2}{32.2} \times \frac{v}{550} = \frac{w\,v^3}{17{,}710}$.

In the case of a one-inch hemp rope (the weight of which may be taken at .33 pounds per foot) running at 150 feet per second, the centrifugal force will absorb:

$$\frac{.33 \times 3{,}375{,}000}{17{,}710} = 62.88 \text{ horse-power.}$$

This gives the point A on the curve—in which it intersects the ordinate representing the velocity selected—150 feet per second. The points in which the curve cuts the other ordinates may be readily ascertained, as from the above formula it is evident that the lengths of each velocity of ordinate intercepted between the curve and the base will be proportioned to the *cube* of the velocity which it represents. Thus $M\,G$ is $= \frac{A\,K}{15^3} \times 14^3 = .0186 \times 2{,}744 = 51$ horse-power. Similarly $N\,H = .0186 \times 13^3 = 40.8$ horse-power; the curve may then be drawn through the series of points thus obtained as shown in the diagram.

For the three-quarter-inch rope the lower curve $O\,L\,F$ is drawn in a manner similar to that just described, and it is only necessary to point out that the starting point of the curve F is obtained by making $\frac{K\,F}{K\,A} = \frac{\text{weight per foot of } \frac{3}{4}\text{-inch rope}}{\text{weight per foot of 1-inch rope}} = \frac{.184}{.33}$, so that $K\,F = .557\ K\,A = 35.5$ horse-power. Any other point in the curve, as W, may be quickly located by making $M\,W = .557\ M\,G$, and by the use of proportional compasses the lower curve can be readily laid down in a few seconds.

A glance at the curve $O\,T\,A$ shows that the horse-power absorbed by centrifugal force in driving ropes at speeds of less than about 30 feet per second is inconsiderable, but that beyond this speed the loss from this cause augments rapidly, until at the point G —140 feet per second, the whole of the allowable tension (taken at the higher value) is absorbed by centrifugal force, while if the lower limit of tension be taken the critical speed is reached at H, corresponding to 130 feet per second.

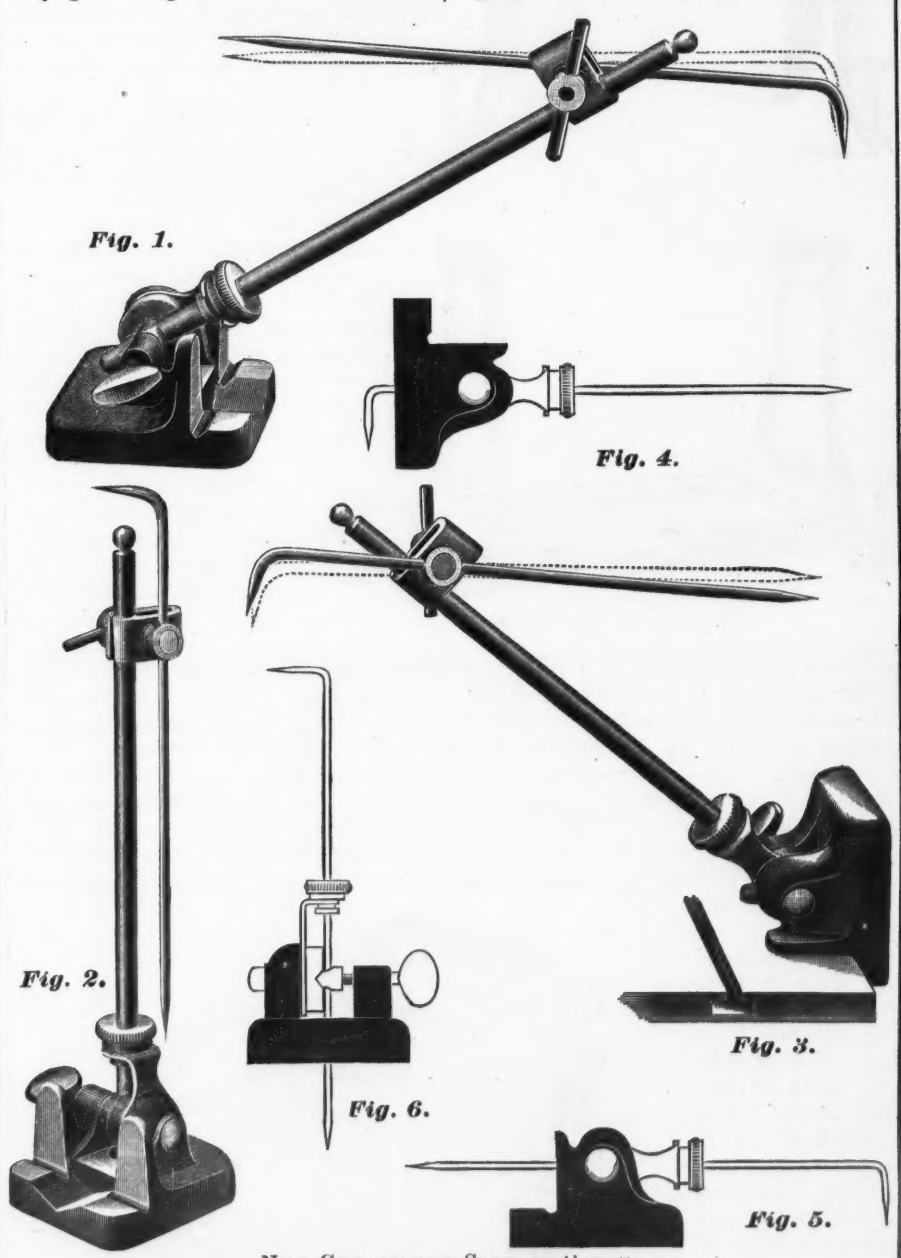
To find the net horse-power transmitted at various speeds by a one-inch rope, and at a given maximum working tension of 200 pounds per rope, it is only necessary to subtract from the several ordinates a length representing the loss due to centrifugal force in each case. Thus at 70 feet per second the apparent horse-power of a one-inch rope is represented by the height of the ordinate $a\,d$. Laying off from d a length $d\,c = a\,b$, we obtain $a\,c$, the length of which referred to the scale $K\,A$ gives the actual horse-power of the rope at the particular speed selected. Similarly, at a velocity of 120 feet per second we have $e\,h - e\,g = e\,f$, and so on for all the other ordinates, the actual horse-power curve $O\,P\,f\,M$ being drawn through the series of points thus obtained. The second curve, $O\ Q\ L\ N$, corresponds to the lower tension assumed for the one-inch rope, while the two lower shaded curves refer to the power transmitted by the three-quarter inch rope at corresponding tensions per square inch. Considering the power curve $O\,P\,f\,M$, it will be observed that the horse-power transmitted attains a maximum at a speed of about 81 feet per second. The precise point at which this occurs may be readily determined by laying off on $A\,K$, from the point B, a distance $B\,x = \frac{1}{3}\,B\,K$, and joining $x\,O$ by a line which cuts the power curve in P, giving the point, and, consequently, the velocity of maximum efficiency of transmission for a one-inch rope, and at the particular stress selected. The corresponding points in the other curves are similarly obtained, and are lettered in the diagram Q, R and S.

Fig. 2 is a diagram of the power of various leather belts per inch of width. Three cases are selected for illustration, viz., single leather, heavy double leather, and a link belt one-inch thick; the back tension is, in all cases, taken as 25 per cent. of the total tension. The following values have been used in drawing these curves:

Description.	Total Stress.	Net Stress.	Weight Per Foot.
Single belt	80 lbs.	60 lbs.	.13 lbs.
Heavy double belt...	150 "	112.5 "	.25 "
One-inch link belt ...	190 "	142.5 "	.48 "

With the detailed instructions which have been given for the construction of the curves in Fig. I, no difficulty should be experienced in drawing horse-power diagrams for belts or ropes of any description. It will be noticed that the link belt—owing to its relatively great weight—attains its maximum power-transmitting efficiency at a much lower speed than the others. The exact speed—determined as in the rope diagram—will be found to be 55 feet per second, at which speed it will, under the conditions assumed, transmit 9.5 horse-power per inch of width. The double belt attains its maximum efficiency under the given stress at a velocity of 67.5 feet per second, and the single belt at 68.2 feet per second respectively, when the weights per foot are as given in the table.

In conclusion, it may be observed that one or two minor points—which are, however, of little but theoretical interest—have not been taken into account in the foregoing. Usually in considerations of the various problems connected with the transmission of power by belts and ropes, a deal of intricate mathematical reasoning is introduced, which, as a rule, complicates the questions at issue without serving any useful purpose. The object of the writer has been to substitute for this a graphic method, which, while possessing the advantage of simplicity, is yet sufficiently accurate for all practical purposes.



NEW COMBINATION SURFACE GAUGE.

Carr's Combination Surface Gauge.

We present with this illustrations of a new surface gauge which is being manufactured by the Hoggson & Pettis Mfg. Co., New Haven, Conn.

It will be perceived that the base is cast and that the stem is attached to it in such a way that it may be inclined to any angle. After being set to any desired angle and approximately to correct height it has a fine adjustment by means of an eccentric washer, this adjustment being available without lessening the rigidity of the stem. The range of adjustment by this means is from $\frac{1}{4}''$ to $\frac{5}{8}''$.

Fig. 3 shows the gauge adjusted to be used at the side of a planer table or other machine tool platen, in which position it can be used for laying out work or for lining up work sidewise, the angle being milled in the upper side of the base for that purpose.

It can also be used in this way at the edge of a bolt slot.

It will be noticed also that there is a V formed in this surface, which shows plainly in Figs. 1 and 2. This V adapts the tool to use upon the edge of a face-plate, so that by it work may be centered upon the plate.

The gauge can also be adjusted to use underneath work, using the upper surface as a guide.

The stem can be removed from the base and the needle substituted, as shown in the smaller views, Fig. 4 of these showing it as arranged for a scratch gauge, Fig. 5 as a scribing block, and Fig. 6 as a depth gauge.

These gauges are made in three sizes, the smallest having two stems, making it a 6" and 9" gauge. The next larger size is 12" and 18", while the largest is 30".

The Hoboken Ferry Company, of New York, contemplates showing at the World's Fair a fac-simile model of the twin screw steamer built in 1805, with original engines and boiler. As a contrast to this venerable craft it will show the model of a modern screw ferryboat, the "Hamburg," built in 1892.

Anti-friction Bearings.

The subject of anti-friction bearings is attracting considerable attention in England as well as here, and a paper on the subject was recently read by Mr. Killingworth Hedges, before the Mechanical Section of the British Association, which is thus reported by *Engineering*.

Mr. Killingworth Hedges next read a paper on "Anti-friction Materials for Bearings Used without Lubricants." After referring to former inventions and practice in this field, the author went on to describe experiments he had made, which led to the introduction of a material which he referred to as carboid. This was composed of finely-powdered carbon mixed with steatite, the mixture having the advantages of plumbago as regards lubricating properties, but showing little signs of wear after tests extending to two years' constant use. Professor Unwin's experiments with the material show that the coefficient of friction is independent of the area of contact, and only increases if the temperature becomes excessive, but not with increased pressure. It tends to diminish with increased speed. The first cost of a dry bearing is less than with brass or patent metal. Besides the economy secured by the absence of lubricants, there is a saving in wages required for oiling and cleaning. The other objections incident to lubrication are, naturally, avoided. Up to the present the chief use of carboid has been for steam-heated rolls. It has also been used experimentally with success for light rolling stock generally.

In the discussion Professor Forbes said that some time ago a Philadelphia company gave up using plumbago alone as a material for bearings, and adopted "fiber graphite." As a *tour de force* a steam engine was constructed in America, in which all rubbing surfaces were fiber graphite, there being no oiled bearing. There had also been spindles running for months at 10,000 revolutions a minute. He had also seen a file working in a graphite bearing as smoothly as if it had been a turned journal. The graphite filled up the cuts in the file and rendered it smooth. He thought that the heating of the bearing described would be considerable, because it was a bad conducting substance. The Philadelphia company found that the fiber graphite would not work if subject to heavy pressure.

Mr. Jeremiah Head pointed out that the bearing, which had been handed round at the meeting, soiled the fingers if rubbed lightly. This showed that the material would wear away, and it would be interesting to know to what extent this was consistent with the claims made by the author. It was well known that if extreme hardness and smoothness could both be insured, a bearing would run without lubrication.

Mr. Hedges in reply said that the longest period his bearing had run was two years, and the wear had been very small. In a pulley bearing on a rope tramway, the wear after three months was only $\frac{1}{10}$ inch. This the engineer to the line had said was a remarkably good result, especially as the pulley was situated on the summit of a hill, where the duty was very trying.

Professor Unwin said that he had had one of these bearings under test, and subject to a considerable range of conditions, some of them very severe. He had crammed the pressure on until the journal had been heated up to purple, but it did not seem to injure the bearing. He did not think that a solid bearing, without lubricant, could be run with so low a coefficient of friction as one with lubricant, but the advantages of doing away with the necessity of oiled bearings were so great that one could well afford some loss of power.

A false statement is going the rounds of the press, to the effect that visitors will not be able to get any drinking water at the World's Fair without paying for it. There will be an abundance of excellent water free to all who want it. Those who wish to drink mineral spring water, piped to the Exposition grounds from Waukesha, Wis., a hundred miles distant, will have to pay one cent a

glass for it. The free water will be that of Lake Michigan, brought by tunnel from a point four miles from shore, and much better than the inhabitants of most large cities are supplied with.

The Bowsher Balancing Way.

In our issue of April 2, 1891, we illustrated a new machine shop tool intended to facilitate the work of balancing pulleys and similar work by substituting for the usual wooden trestles and straight-edges, which must be carefully placed in position, and leveled for each job, a tool which, when once in place and leveled up, would answer for all work of the kind, without readjustment until it became necessary to move it to another part of the shop, when its readjustment or leveling takes but a few seconds, the levels being permanently fixed to the tool, and two screws in the base serving to level it upon its three points of support The present illustration shows the device in its latest form. The surfaces upon which the arbor rests are chilled and ground true, one standard being movable upon Vs the same as the foot-stock of a lathe, to suit varying lengths of arbors. Since our former illustration of the device, it has found favor in a large number of the best shops where much balancing is done. It is made in three sizes by N. P. Bowsher, South Bend, Ind.

Combined Punch and Shear.

We present herewith an illustration of a combined punch and shearing machine which is arranged to be driven by power, and so that shearing of flat or bar iron, round iron, or punching can be done without changes in the machine and without interference between the various functions.

The frame of the machine is double and symmetrical, the lever, eccentric and slide being placed between the two sides, all the boxes for the shafts being cast solid with the frame.

There are three sets of knives: one for round iron and two for flat iron, these being independent of each other and of the punch. Any one of these may be used alone, or all may be used together, as desired, and thus there is no time lost in making changes. All shafts, bolts and plungers are of steel, and all the knives are fitted into solid castings. There is a clutch worked by a hand lever which brings the machine under control. Improved gauges and strippers are provided.

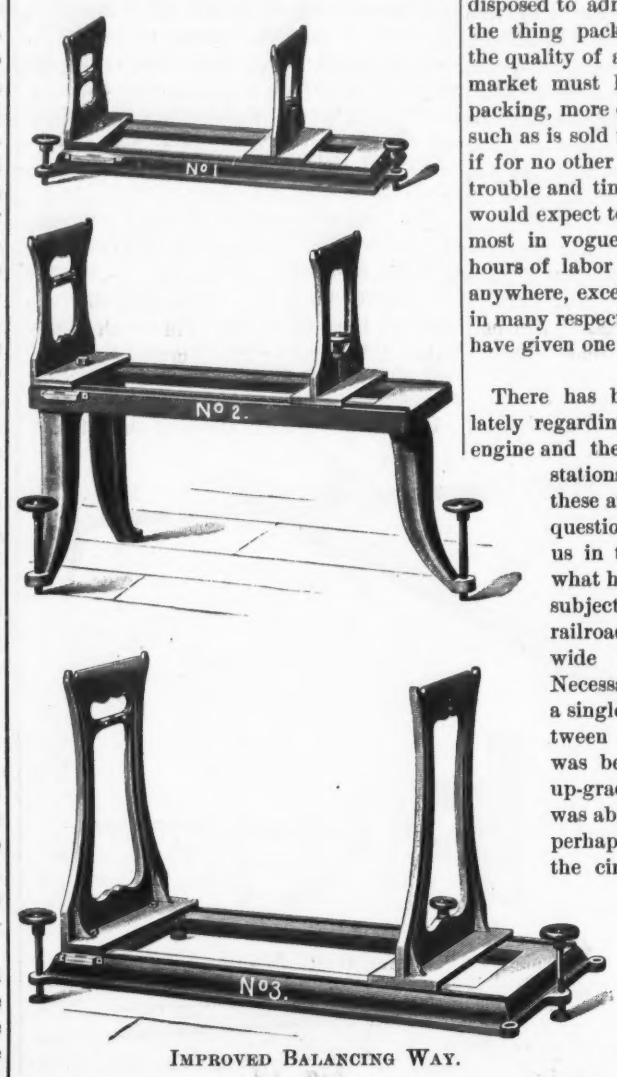
The machine is made in ten sizes, that illustrated being the smallest and having a capacity to punch a $\frac{1}{2}''$ hole in $\frac{5}{16}''$ iron, to cut by one pair of shears $4''$x$\frac{5}{16}''$ iron; by the other $\frac{1}{2}''$x$1\frac{1}{2}''$, and by the third $\frac{3}{4}''$ round. It is built by the New Doty Mfg. Co., Janesville, Wis.

Packing Goods—Water Capacity of Boilers.

BY W. H. BOOTH.

There are other considerations than mere excellence which go to selling a tool or other appliance, and one of these is the method by which it is packed.

The old method of packing goods of all kinds in the hardware line, as practiced so long in England, was to fold them in brown paper, and when sold in a retail store there was, of course, necessarily an amount of untying of string and unfolding to be done, and the counter was strewn with paper, and general untidiness prevailed, not to mention the time wasted. The same articles packed in card boxes could be examined in a tenth the time, and it is some years ago since I first drew attention to this fact, as I believe it was largely to the better and more convenient packing of many American articles that they were indebted for their first sales, and if at all satisfactory, repeat orders followed. No doubt the cheapness of timber in America enabled many other goods to be better packed than they had been by English makers, and the latter lost a lot of business entirely on this account, though in this respect there has been an improvement. For a standard article there is call for design in the box in which it is packed, quite as much as for the article itself, and a suitable box has no surplus dimensions; it holds the article firmly without shake, and according

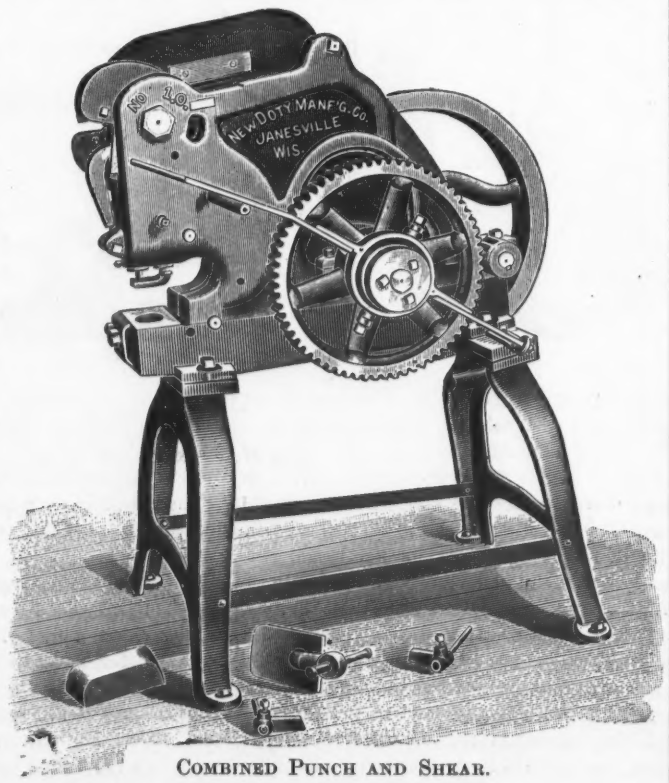


IMPROVED BALANCING WAY.

as the case is meant merely as a protection during carriage or to serve as a permanent box, securing cleats will be fixed by nails or arranged to swivel to fix the article immovably in place. I am sorry to say there is on this side still too much use of wood shavings to pack articles which would perfectly well be packed without. I do not refer merely to their use in packing odd castings from the foundry, but finished articles of standard frequent sale, which could as well be packed clean. The salesman in a store, even if like our friend "Anti-brag," stuffed full of "patriotism," will run a bit dry in that direction, and insensibly tend to show to a customer those goods most easily got at and put away again, and I have observed both of American and German imported goods that the most successful firms are those who pay attention to packing. The shape of a box, cardboard or otherwise, need not necessarily be rectangular, but the depth should be equal throughout for piling purpose. In plan a case may be square, rectangular, trapezoidal, or even triangular, for in no case would space be thrown away, as two cases end for end of any of the above shapes will make a rectangle.

Whatever is done, extraneous material for filling or padding should be avoided, and in this respect American packing is undoubtedly at the top, though to a very much less degree than was the case only a few years ago, for our own manufacturers are now displaying far greater care in this important item, on account, of course, of the competition they have met with outside. The excellence of American packing is cheerfully admitted by men who are by no means disposed to admit any special excellence in the thing packed. Anyhow, whatever be the quality of an article, its success in the market must largely depend on mere packing, more especially if the article be such as is sold retail and over a counter, if for no other reason than the saving in trouble and time. Of all countries, one would expect to find time-saving methods most in vogue in England, where the hours of labor are shorter than they are anywhere, except in Australasia, and yet in many respects this is not the case, and I have given one instance above.

There has been considerable discussion lately regarding the relative power of the engine and the boiler in power generating stations of electric railroads, and as these are so rapidly extending, the question will come home to many of us in the near future. The gist of what has been so far written on this subject is to the effect that in small railroads there is likely to be a very wide range of power required. Necessarily upon a road running but a single car, the load would vary between a maximum, when the car was being started on the steepest up-grade, and a minimum, when it was absorbing no power at all, but perhaps even putting some back into the circuit. With two cars there would still be occasionally an equal range of work, and so on, for small numbers of cars, the variation is always likely to be great, though the maximum effort will be more infrequent. Over a large area with a hundred cars at work, there might be little or no variation. One thing is very clear, there must be power enough in the engine to overcome the maximum resistance without serious diminution in speed of the cars, and the question naturally presents itself thus: Given a maximum power of say 100, and a mean of 60, what ought to be the economical rate of power for the engine to be employed?



COMBINED PUNCH AND SHEAR.

The minimum power can never be nil, for when all the cars are stopped the engine has itself to turn, and the minimum may be say 10. If the engine be of 80 power for its best economy, it will be very seriously underloaded when only doing 10, and overloaded when at the maximum of 100, but the maximum rate is shown by diagrams, taken by electricians, of the current needed, to be merely momentary, and yet there is a good deal of time spent at very low powers, and we know how very uneconomical an unloaded engine is, whereas an overloaded engine is not so seriously wasteful, and it appears probable that, if the maximum power can just be secured by an almost full stroke admission of steam to the cylinder, there will be a better prospect of economy than when the engine is large enough to be fairly

economical at the maximum rate of working, for, if made too big, there is so large a steam consumption in merely turning the engine itself at lower powers, that the high economy of large power running is swamped. It is important that engineers in charge of power stations should endeavor to compare the total indicated power of the engine with the work done upon the driving belt, for then only can the true proportions be studied. When it comes to the question of boilers the case is very different, and the errors of too great an allowance of boiler power are clearly seen upon a little study. Like a fly-wheel which absorbs and gives out power by a variation in its velocity, so does a boiler by a variation in its contained pressure, which sets free or absorbs heat, and so serves to equalize the variable power of the engine, and distribute it evenly over a longer period of the boiler's working.

Thus in the figure the line *A B* may represent the mean rate of doing work. The peaked line *c d* may show of what extreme variations this mean is built up, whilst the wave line *e f* will serve to show the variation of the demand upon the furnace of the boiler, the peaks of the duty diagram *c d* being toned down by the regulating capacity of the water contained in the boiler. The larger the quantity of water in a boiler the more closely will the wave line *e f* coincide with the mean *A B*, whereas a boiler with small water capacity might show a diagram of demand upon the furnace, as per the dotted line *K K*.

Any one can go into the figures for himself, and will perceive that to equalize the stress of firing there is nothing so important as a good capacity for water. Steam space is by no means specially important, except that it is often an aid to the prevention of priming. It is water capacity that must be secured if the furnace is to be steadily worked, and a steady equable rate of furnace work is one of the most important factors of economical working. Every reduction in water capacity means an increase in the variation of the demand made upon the energies of the fireman, and a man who would easily keep going 100 square feet of grate surface under 40 tons of water would require an assistant if his boilers held only 15 or 20 tons, simply because the diagram of demand upon the man would become more peaked, and the highest peak would perhaps overtop his maximum output. It follows as a corollary that where a steady amount of steam is demanded, as in the driving of a large textile factory, the water capacity of the boilers is a matter of far less importance; that is the case with small establishments having a variable demand for steam. The small water capacity boiler has its place, but it is in the fire engine, where rapid steam raising is more important than steady steaming power, this latter being endeavored to be obtained by the use of such fuel as will keep up the pressure to blow-off point all the time. Fuel economy is not sought in the fire engine itself, as the fuel to be economical of lies at the far end of the water jet thrown by the engine. Then the duty of the fire engine is a steady duty, and once at work the feed and fuel are supplied fairly continuously. It may fairly be ruled that boilers should hold water proportional to the range of work of the engines which they supply with steam, and it may likewise

be stated that where the demand for steam is unsteady there is a marked advantage in having as large an area as possible from which steam is evolved, in order that water may not be entrained along with the steam.

Cam Movements—II.

BY A. D. PENTZ.

In designing a cam to oscillate a lever at a high speed, one-third of the time of revolution of cam to be consumed in the outthrow and two-thirds in the inthrow, construct the diagram *B*, in Fig. 5. The circle through *E* indicates the point inward and that through *D* the point outward beyond which the center of the roller must not go. Indicate the point near *D*, at the intersection of a line in the web on the outer circumference of the throw, 120 degrees forward from *D*, and at the intersection of the circumference indicate the point *C*. Then divide the web line that is situated half way between *D* and *C*, or the 60th degree, into as many parts as there are web lines that cross the outthrow, or 12 divisions, and number them as shown. Now with dividers set to distance from *A* to 1, and with a center from which its arc will pass through *D*, and through 1, indicate a point on the web line 10. Then with the dividers set to *A*, 2, and with a center that passes its arc through the point 2 and through the new point on the web line 10, indicate a point on the web line 20. Again shorten to *A*, 3; find the center that passes the arc through 3 and the new point on 20, then indicate a new point on 30. Repeat this method until the line 50 is reached. The middle line, 60, is always bisected by the cam curve under this rule.

The points for the outer half of the outthrow being found, set the dividers to *A*, 11; then from the center that passes the arc through *C*, and through 11, indicate a new point on the line 110. Next with the dividers set to *A*, 10; on an arc that cuts 10, and the last made point on the line 110, indicate a new point on the line 100. Repeat this, outward to the central web line, and when completed connect the point *C*, through the points thus indicated, to the point *D*, by a smooth and continuous curve. The inthrow curve is made by the same method of spacing the central web line. But in this case this curve is so long and gentle that only every second web line need be indicated by a point.

In some cams the position of the roller is an arbitrary one at each point of revolution. To design such a cam it is only necessary to indicate these arbitrary points and to draw a continuous curve through them. In a large number of cases, however, the roller has arbitrary positions at a limited number of points only in the revolution, in which case the diagram is filled in as in Fig. 6. In this case the circles *D* and *A* indicate the extreme throw of the cam and the arcs *B* and *C* indicate intermediate reversals and movements. Starting at *D*, the full outthrow is had in 120 degrees. The first inthrow is had in 80 degrees, then a reversal is had in 80 degrees, after which the final outthrow is had in 80 degrees. While the throw between 180 and 220 degrees is very rapid, this cam properly made and not overloaded need not be noisy at any speed reaching to 10,000 inches per minute.

Cams cut within the convex faces of cylinders such as is shown in Fig. 7 are very satisfactory when well made. This figure indicates the movement of a sewing machine thread take-up. The casting *G* is the arm within which the shaft that operates the cam *B*, *A*, is mounted. The lever *C* is pivoted to the same arm by the screw stud *D*. This lever is operated by a roller on the stud *E*, and it controls the thread within the hole *F*. Cams of this character sometimes run 3,000 revolutions per minute. Such a cam, to be at all satisfactory, must be cut on a special machine in which the cutter must oscillate about a pivot which has its center in the same exact relations to the cam as the fulcrum of the driven lever has on the machine in which it is used.

Fig. 8 shows two diagrams. These diagrams are the convex surfaces of the cylinders laid out flat, and they have web lines on their surfaces to correspond with each 10 degrees of revolution of the cam The one with straight web lines is that upon which a

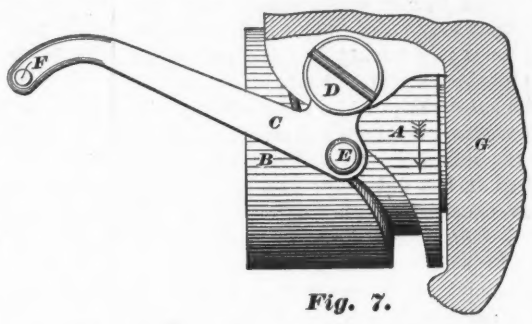


Fig. 7.

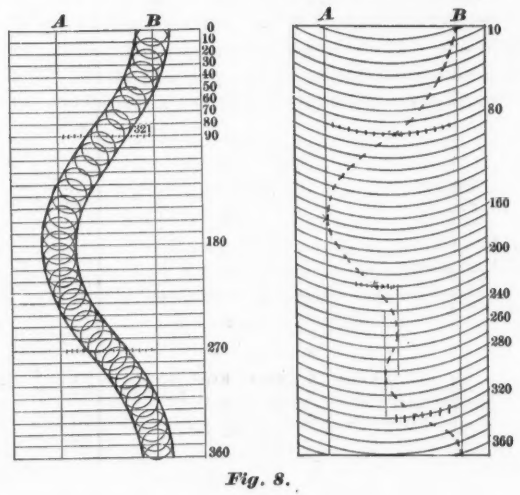


Fig. 8.

centers, circles as large as the cam roller are drawn, and these circles are joined by a tangential curve to show the flanks of the completed cam. These flanks frequently vary considerably from the center line in shape, but they must not be confounded with the center line, because that alone is the cam movement. The diagram in Fig. 8, with arc web lines, is intended for the design of a cam which is to oscillate a lever that is provided with a roller whose center moves within the exact arc shown. The cam is to be laid out on this diagram by the rules already given in the descriptions of Figs. 5, 6, and the other diagram in Fig. 8.

In such cams as shown in Fig. 7, the center of motion *D* of the lever should not be at a distance above the center of the cam equal to the length of the arm *D*, *E*. The reason of this is illustrated in Fig. 9, in which *A* indicates the position of the fulcrum of the lever

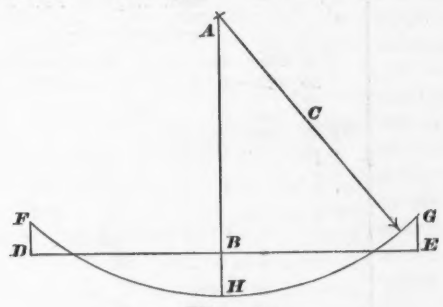


Fig. 9.

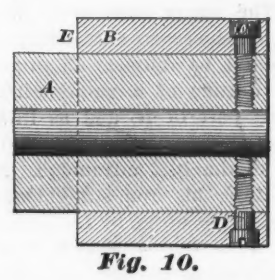


Fig. 10.

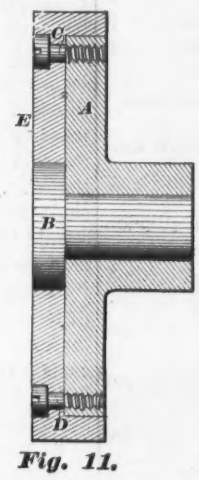


Fig. 11.

one flank is transferred from the diagram and then the hub is driven out. This permits of drilling and filing of a satisfactory kind to be done. The screws *C* and *D* are placed where they will not be cut into, and by their use the sleeve is always returned to its proper place on the hub. This sleeve should be fitted to the roller in position on its lever and in the machine so that all the angles and curves shall have their correct inclinations. If a cam-cutting machine is to be made this one side is a sufficient guide from which to make the former, and if the shape need experimentally to be tried, a spring on the lever will cause the roller to follow satisfactorily for a time. Should, however, a solid cam be required, another sleeve may be fitted on and the other flank of the groove cut into it. It thus may be fitted perfectly, so that the roller will move freely and not shake within its groove.

Fig. 11 indicates how a cam may thus be worked in the end face of a cylinder. The cap *B* is removable from the flange *A*, to facilitate the cutting of the outer flank of the groove after it is laid out on the face *E*.

The getting of the shape of a cam like that in Fig. 7 must for some purposes be done experimentally. No one can exactly calculate the thread take-up cam for a sewing machine. They are best designed by using a machine complete, except that there must be no roller in the lever and that the cam must be a blank. The roller hole in the lever should be drilled out the full diamter of the roller. The machine is threaded up as for sewing very heavy work. Then it is turned from one point to another and the take-up lever is adjusted to control the thread just right at each of these points, and when satisfactory the circle of the cam roller hole in the lever is scratched through upon the convex surface of the blank cam. The assumption that a cam roller for this kind of cam should be a conical frustum is neither true in theory nor does this form wear well in fact. It is true that a cylindrical roller is not theoretically correct for a cam cut within the convex side of a cylinder, but in practice it is found quite satisfactory. Such rollers, however, if made in quantities, for economical assembling in machines, may profitably be crowned one per cent. of the length of the face.

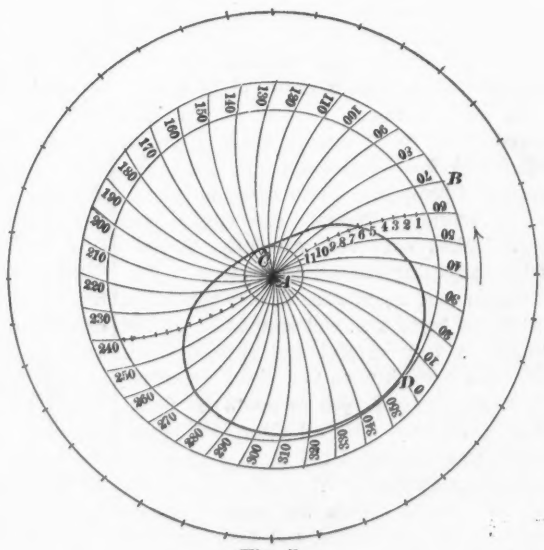


Fig. 5.

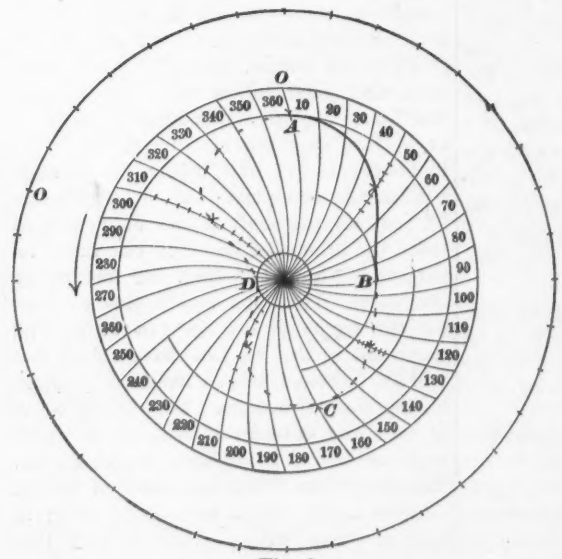


Fig. 6.

CAM MOVEMENTS.

It is Patented.

We are requested by Messrs. Bardons & Oliver, of Cleveland, O., to state that the forming tool slide which we illustrated in our September 1st issue, as put on the market by them, is a patented tool. We omitted mention of this fact in our illustration, believing that mechanical readers would have no trouble in drawing the correct inference in the matter, which is that tools and machinery described as being put on the market by machinery establishments as a part of their regular line are always presumably protected by one or more patents wherever possible to so protect them.

Some experiments have been made at Liége, Belgium, in forming alloys by compressing together pulverized copper and zinc in different proportions, but nothing of importance has so far been developed. The pulverizing seems to have been done by filing.

The Pennsylvania Railroad is laying a fifth track between New York and Philadelphia.

LETTERS FROM PRACTICAL MEN.

Standard Threads.

Editor American Machinist:

In your issue of Aug. 18 you give an excellent illustration and description of a new lathe by the Shellenback Machine Tool Co., which possesses special facilities for cutting threads. I regret to notice, however, that 13 threads to the inch is a pitch not included in the list and as this is the standard pitch of a half-inch bolt which is largely, and should be uniformly adopted, I think all lathes intended for cutting threads as fine as this should include that number, as they should all the U. S. Standard pitches within the range of the lathe. These pitches are as follows: 2¼, 2⅜, 2½, 2⅝, 2¾, 2⅞, 3, 3¼, 3½, 4, 4½, 5, 5½, 6, 7, 8, 9, 10, 11, 12, 13, 14, 16, 18 and 20 threads to the inch.

Any lathe having a capacity to cut threads from 20 to the inch, to say 3 to the inch, should be capable of cutting any number in the list between these two, while larger lathes should include all the fractional numbers between 2 and 6.

One of the most effectual bars to the use of standard screw threads that I can conceive of is to have the machine shops of the country largely equipped with lathes which will not cut such screws, and as the use of standard screw threads is of the greatest advantage to manufacturers, it would seem as though those of them who make lathes should pay attention to this matter. The lathe referred to is by no means the only one which might be similarly criticised, and it could, of course, be easily provided with gears to overcome the difficulty.

S. T. FREELAND.

Areas of Cylinders for Compound Engines.

Editor American Machinist:

Referring to the criticism of my matter on compound engines in your issue of August 25th, permit me to state that the method adopted by me is that of developing the several sizes or rather areas of cylinders from the mean pressures, and to insure equal loads and work on the pistons. There are four expansions provided for in the several cylinders of the compound example, and with the 33 pounds initial pressure for the low pressure cylinders we have, as shown, 19.687 pounds mean pressure, or a mean effective pressure of 15.687 pounds. Comparing this 15 687 of the low-pressure cylinder with the 53.11 of the high-pressure cylinder, we have a required ratio of areas between the two cylinders of 1 for the high-pressure cylinder to 3.385 for the low-pressure cylinder. The steam in the single-cylinder engine has 20 expansions, and that in the compound has $5 \times 4 = 20$, or 5 expansions in the high and 4 in the low-pressure cylinders.

There is no necessity for developing theoretical cards or diagrams that require to be modified to provide for equal work on the cylinders.

WM. H. WEIGHTMAN.

German vs. American Machinery.

Editor American Machinist:

In your issue of August 11th, just at hand, I notice a letter from Mr. W. H. Booth criticising my statement that American tools do not at present have to take a back seat to anything produced in Germany.

My remarks were not directed in any way against Mr. Booth, but rather against the "American Engineer" who made such sweeping statements as "The German work *is better* than anything produced in America," and "The German tools are *not equaled* by anything either in America or England."

Of course I may be mistaken in my identity of this gentleman and the firm he represents (which is, however, of little moment), though I very much doubt it, as, although London is a big town, there are not many American engineers located there selling German machine tools.

With all due deference to Mr. Booth (who probably never experienced the difficulty of selling German machinery in England), I must adhere to my statement, that there is still a very strong prejudice against it, the prevailing idea being that all things German are "cheap and nasty."

The prejudice that at one time existed against American tools is rapidly disappearing, as evinced by the large trade done by Messrs. Churchill & Co., Buck & Hickman, and many others.

In our own factory, though not at present very extensive, we have several American lathes, two Brainard mills, two Lodge & Davis drills, and a variety of small tools by Brown & Sharpe and other makers, and in three other shops closely connected with our own there are machine tools by Brown & Sharpe, Pratt & Whitney, Lodge & Davis, Flather, Barnes, Garvin, Dwight Slate, etc., but I do not know a single instance of German tools being used.

Though I do not care to enter into a political discussion with Mr. Booth, I inclose my card, and shall be very pleased to show him our American tools at work whenever he may be able to spare time to favor us with a visit.

ANTIBRAG,
An English Fair Trader.

Turning Shafting—High-tempered Bearings—Turning Locomotive Tire.

Editor American Machinist:

"Passaic's" letter in your issue of Dec. 31, 1891, has not received the attention among your readers that I had hoped it would. It will be remembered at the time his letter was written that 10 feet in three days was the rate that he could turn shafting, and keep the quality up to his standard. It appears to me that for the smaller sizes he could use a process shafting that would certainly cost much less, and in all respects be as round and straight as he could turn, file and lap. The larger sizes, of course, must be turned, but I see no reason why so much as ¼" should be roughed off to get the size. The outside of a wrought-iron shaft is the best part of it, and as little should be turned off as possible, if the shaft is to be worked to the best advantage. If "Passaic" will order his iron straightened (if he is not prepared to do so) by the manufacturer, he will find that ⅛" will be sufficient to allow for turning any size of rolled iron. Large forged shafts might require more in some instances to true them up.

To turn anything round, the tool must have a keen cutting edge, and as long as it remains so there will not be much trouble in turning the shaft true to size and round, implying, of course, that we have a good lathe and good follow rest. The trouble that I have experienced in turning shafting, or, in fact, anything else, begins when the tool begins to get dull, before the carriage reaches the face-plate. As the edge wears off the tool, the shaft is not only enlarged twice the amount of the wear, but more resistance is required to keep the tool up to the work. Then the tool begins to hunt for the seams or soft spots in the bar, and the result is that we shall have a shaft that will not be round, but an irregular polygon having many diameters. The cutting edges of taps and reamers apparently, at least, wear much longer than lathe or planer tools. Reasoning from this, I have thought that an external reamer might be a good thing to turn shafting with. Another plan that might work well would be to substitute an emery wheel for the tools to take a finishing cut. The wheel would maintain practically the same diameter while the shaft was being ground, and as the resistance required to keep it up to work would be very slight, there would be practically no spring, and in consequence we should have a round shaft, having but one diameter. No doubt these or other and better schemes have suggested themselves to "Passaic," and should he have put any of them into practice I will, with the editor's permission, invite him to contribute his experience to these columns.

One who has been troubled with "high-tempered" bearings will naturally find Bell Crank's paper in AMERICAN MACHINIST of June 16 very interesting reading. A remedy would have been quite as welcome as the diagnosis was interesting, for "high-tempered" journals are not so rare an occurrence as Bell Crank may imagine. I beg to assure him that it would be a tenderfoot mechanic who would question the truthfulness of his statements. I know how discouraging it is to have charge of an engine that has been designed and built with the greatest care, and then have it run hot, or, what is worse, insist on being heard when a golden silence would be worth diamonds. Then if one wishes to make discomfort complete, all he has to do is to go over to Swashbuckler's and see his engines, that were neither designed nor built with care, running cool and quiet. If there is one thing that is surer than an other, it is that some bearings will run hot as long as they are kept above water, or till there is some radical change made either in construction or the material in which the shaft runs. I have known brass substituted for babbitt metal with good results, and in other cases babbitt metal proved to run cool where brass would not. Such experience perhaps has led to sandwiching the two metals.

I would like to relate one instance right here that may be of interest. Several years ago the engine driving the dynamos that had run for about a year without the least trouble, got a hot crank-pin one night about 3 A. M. The engineer staid out and reported the case in the morning. He did not know what caused it, but did know that he had oiled it as usual. The pin was cut on one side only; both boxes, however, were ground out about the same amount. I had the engine examined all over without finding any cause for the hot box, so concluded that it was neglect on the part of the engineer, and told him so. I had the pin turned, the boxes parted and bored out to fit the pin, and all put together ready for the engineer to start

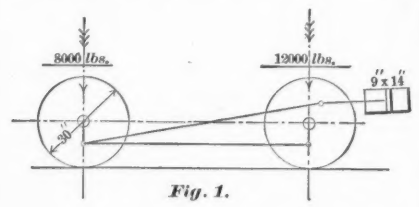


Fig. 1.

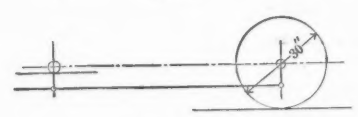


Fig. 2.

up when he came out in the evening. When he started up the pin began to heat, and he sent the watchman up, who informed me that the pin was as hot as the night before. I did not go down, but sent word to put plenty of oil on it, and it would get cool before morning. But it did not, and in the morning I found the boxes and pin cut up, as the morning before. The engine was again looked over very carefully, without finding anything but what was O. K. The crank-pin was again turned, and the boxes parted and bored out, and scraped to fit the pin, by a very careful and skillful man. This time it was known to a dead certainty that it was as good a job as could be done. That the oil holes were all clear and clean. That it was not keyed too tight. That the eccentric was properly set on the shaft, and had not moved. That the valve had not moved on the valve stem, and that it opened each port alike or nearly so. That the crank-pin was parallel with the shaft, and that the shaft was at an angle of 90° with a center line through the cylinder, and was level. There was no springing to get the parts together. The liners that were put behind the boxes to fill up the strap to make up for that cut, and bored out, were shaped to the exact thickness. Now when this engine was started up the crank-pin got hot, and nothing but water would keep it cool enough to run. The next day I took the dimensions for a new pin and boxes, and had the old pin turned, and the old boxes filled with babbitt metal, and the engine put together, thinking that we might get one night's run out of it while the new pin and boxes were being made. It was something of a surprise, but the boxes and pin that I had made were not used, for the pin did not get hot afterward, or give any trouble in the least. I have given up trying to find a cause for such phenomena. If brass proves troublesome, I try babbitt, and *vice versa*. The engine is medium high speed, 12"x20", improved plain slide valve type, built by a western firm, whose engines I have since learned are liable to run hot, especially the crank-pin.

W. E. Crane, in AMERICAN MACHINIST of June 23, has touched a part of the locomotive that will bear being written about. I refer to the unequal wear of tire. The engine Mr. C. speaks of had 5-foot wheels varying ⅛" in diameter, which averages pretty closely to my experience. Of the several locomotives we have, I will call the reader's attention to one that is partly represented by Fig. 1. When this engine came into the shop to have the tire turned for the first time, it was found that the forward wheels were about ⅛" smaller than the back wheels were. In scheming for a cause for the unequal wear, the mate to the engine was weighed, and with about half tank of water, to get average weight, the weight on each pair of drivers was as shown in the sketch, viz., 1,200 pounds on the forward, and 8,000 pounds on the back driver. This, in one sense, accounts for the forward wheels being the smallest, owing to the greater weight they have to carry; for the greater the weight, other things being equal, the faster the wear, while on the other hand the back driver should slip, owing to the lesser weight that they have to carry, and thereby wear faster than the wheels that stuck to the rails. I should have stated before this that, so far as could be seen, the metal of all the tires were of a like hardness. There can be little doubt but that the slipping will take place between rail and back driver, while it is equally true that the extra weight is more destructive to the tire than the slipping. In order to have an average diameter, I have the back wheels turned about $\frac{3}{16}$" smaller than the forward pair. On the other style engines the variation is not so great, but in turning, the diameter is averaged on all that show a disposition to unequal wear. I am certain that the pulling capacity is not appreciably affected by a variation of say ¼" in 5-foot wheels. Thinking this statement may be doubted, I will inclose Fig. 2, showing an extreme case of variation in the size of wheels, from which an idea may be more readily obtained than from sizes that we may expect in practice. We will call the drivers 3" and 30" in diameter, and will suppose that sufficient weight is applied to them to prevent slipping of both pair, for it is apparent that as the wheels are connected, if the engine moves one pair must slip, and also if there is anywhere near the same weight applied to each pair, that it must be the big wheels that will do the slipping. The extent of their slipping will be the difference in the circumference of the wheels at each revolution. The power lost would be the force required to slip these wheels the distance just stated, but the amount the engine would pull, regardless of speed, assuming the same pressure to be applied to each crank-pin, would be equal to an engine having all wheels the average diameter of Fig. 2, viz., 16½". Hence I think it can safely be said that, so far as the pulling capacity of an engine is concerned, the more the wheels wear down, whether equally or not, the more the engine will pull, provided, of course, that the weight on the wheels shall be sufficient to prevent the slipping of but one pair on engines having but two pairs coupled.

The next step will be to determine approximately the power lost by having coupled wheels of different diameters. As easy a way to do this as any is by an example, as follows: Take an engine of the "8-wheel American" type, having the small wheel of such diameter (5.6 feet) as will require 300 revolutions to the mile, and the big wheel 1" larger in circumference. It will be seen that in running one mile the big wheels must slip 300", or 25 feet.

The ratio of the distance slipped by the big wheels to the distance traveled will be 25 to 5,280. To see how much this slip amounts to, it will be assumed that the engine runs at the rate of a mile in a minute,

that the cylinders are 18″x24″, and an average effective pressure on the pistons is 50 pounds per square inch. The pull of this engine will then be $\frac{18^2 \times 2 \times 50}{5.6} = 5{,}785$ pounds. The H. P. will be $\frac{5{,}785 \times 5{,}280}{33{,}000}$ = 925 H. P. Now if the wheels had all been the size of the large ones, the engine would have traveled 25 feet further; hence the power required to move the load 25 feet is lost, which will be $\frac{25 \times 5{,}785}{33{,}000}$ = about 5 H. P., or say a half of 1 per cent. A better method perhaps to calculate the power lost will be to assume a weight for the drivers to carry, and a force that, if applied to their circumference, would slip them. The weight on a pair of wheels for such engine will be about 27,000 pounds, and the force required to slip them ought to be just half what the engine will pull, which is usually one-fourth of the weight on the driver. Hence the force required to slip these wheels will be 27,000 ÷ 4 = 6,750 pounds. This amount multiplied by the 25 feet slipped, and divided by the constant 33,000, will still give very nearly 5 H. P. as the amount lost. Of course the percentage of power lost would increase as the pressure used decreased, but 50 pounds average pressure is not high, except for very high speed, so I feel safe in saying that the power lost in the practical difference in the diameter of locomotive drivers cannot be detected.

As for a single pair of drivers, the tendency out west is to increase the number to three pairs for heavy passenger traffic.

C. O. SINE.

A Study of Fly-wheels—III.

BY LEICESTER ALLEN.

We come now to the case in which both the driving power and the resistance vary during a single turn of the fly-wheel, and in which these variations are not equal and synchronous.

As good an example of this case as can be selected is that of an air compressor driven by a steam engine working expansively, the piston of the engine and that of the compressor being joined to a single piston rod; or, what will practically produce the same relations of stroke, connected by rods of equal length to a common crank-shaft with cranks of equal radius, set either together or 180° apart. The strokes will thus be equal, and any position of the steam piston during its stroke will exactly correspond to that occupied by the piston of the air compressor.

Let us suppose, as an example, that a compressor whose piston diameter is 14¼″ and 28″ stroke is to be driven by a steam engine, taking steam at 80 pounds gauge pressure and making the same stroke; the piston of the compressor to be connected to a crank set at 180° from the crank of the engine, and to be used for compressing air adiabatically from ordinary pressure to 44.07 pounds total. Let us further assume that a clearance of $\frac{1}{10}$″ is allowed between the heads of the compressor cylinder and the piston, and that the outlet and inlet valves of the compressor are of the puppet variety, with beveled seats, and seating flush with the inner surface of the compressor heads.

With reference to the engine that drives the compressor, let us suppose that, in order to compress the exhaust steam enough to fill its clearances with steam at initial pressure, it is necessary to close the exhaust at 1⅛′ from the end of the stroke, and that the cut-off valve is to be set to cut off the steam at 7″ of the stroke. The number of turns is to be 72 per minute, and the friction of the machine is assumed to be ten per cent. of the piston work in both the engine and the compressor.

We are to design a fly-wheel for this machine, that, when the specified speed of running is maintained, shall vary in velocity not more than 6 per cent.

Having constructed a theoretical diagram for adiabatic compression of air from ordinary atmospheric pressure (14.69 pounds) to a total of 44 pounds, or to 29.38 pounds by gauge, as shown by the outline *e f m h g*, Fig. 1, with due allowance for the resistance of the valves, we find the mean effective pressure per square inch is 19.3 pounds. This, multiplied by the piston area in inches and length of the stroke in feet, gives us the useful work in foot-pounds. As the clearance is $\frac{1}{10}$ inch, the air in this clearance at 44.07 pounds total pressure will expand when the stroke is reversed to fill a space $\frac{3}{10}$ inch, so that the effective clearance is $\frac{3}{10}$ inch, and the effective stroke is 27.7 inches, or 2.3 feet. We have for the useful work per stroke 171 × 19.3 × 2⅓ = 7,700.7 foot-pounds. The scale of stroke line is ⅓″ = 1″, and the scale of pressures is $\frac{1}{8\frac{1}{3}}$″ = 1 pound.

To the useful work must be added 10 per cent. for friction, or 770.07 foot-pounds, making the work of the compressor equal 8,470.77 foot-pounds. This is the work the engine must perform in addition to its own friction under load assumed in our example as 10 per cent. It follows that 8,470.77 foot-pounds is 90 per cent. of the total work of the engine, and that the latter is 9,470 foot-pounds.

Now if we divide the total work of friction (which is the total work of the engine minus the useful work of the compressor = 9,470 — 7,700.7 = 1,769.3 foot pounds) by the product of the area of the compressor piston in inches, multiplied by the stroke in feet, we shall get the pressure of the friction on a square inch of the piston.

$\frac{1{,}769.3}{171 \times 2\frac{1}{3}} = 4.4343$ pounds.

Setting off this pressure in Fig. 1 below the atmospheric line, and indicating it as constant by the line *c d*, the area *c d m h g* represents the useful work plus the work of friction performed on each square inch of the compressor piston by the steam piston.

We now proceed to construct our diagram, Fig. 2, for steam inducted at 80 pounds, and cut off at quarter stroke. In constructing this diagram I have used the exponent of Grashof for steam containing 10 per cent. of water, which gives an expansion line sufficiently near the average of steam engines for the present purpose.

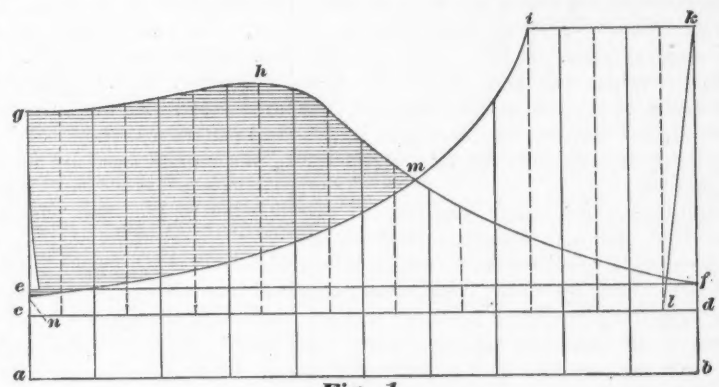


Fig. 1.

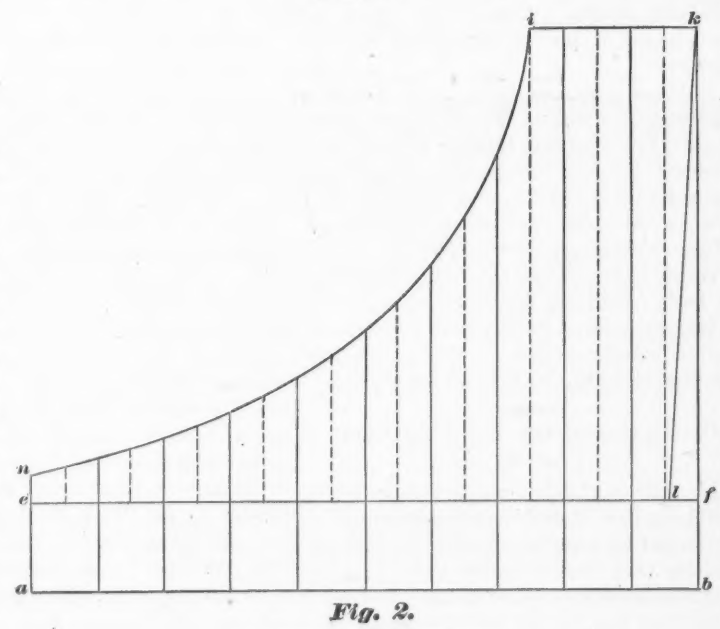


Fig. 2.

A STUDY OF FLY-WHEELS.

The diagram is shown in Fig. 2, in which *i k* is the line of induction; *i n* the expansion line; *e l* the atmospheric pressure line; *a b* the vacuum line, and *l k* the compression line. The area *l k i n e* indicates the work performed by the steam on each square inch of the steam piston, and by averaging the dotted ordinates the mean effective pressure is found to be 39.1 pounds per square inch.

Dividing the total work by the mean effective steam pressure in pounds, multiplied by the stroke in feet, we find $\frac{9{,}470}{39.1 \times 2\frac{1}{3}} =$, 140.66 as the required area of the steam piston and the nearest convenient diameter is 13⅜′, which is the diameter of the steam piston.

Now the mean effective pressure plus the friction pressure indicated on the air compressor diagram is 19.3 + 4.4343 = 23.7343 pounds, and if we superimpose upon the air diagram another, in which the ordinates representing the same points of stroke as the ordinates in the steam diagram are each obtained, by multiplying the steam ordinates by $\frac{23.7343}{39.1}$, and draw a curve connecting these ordinates, we shall obtain a diagram *l k i m e c*, the part *k i m e c* of which indicates at any of its points the pressure per square inch that the steam exerts on the compressor piston plus the pressure due to frictional resistance. The base line of this diagram must be the friction line *c d*.

The ratio $\frac{23.7343}{39.1}$ reduces to 0.6; therefore each ordinate of the curve should be 0.6 of the corresponding ordinate of the curve for steam in Fig. 2. The total resistance overmasters the driving power at the point *m* in Fig. 1, and the shaded area *m h g e* indicates the work in foot-pounds that the fly-wheel must supply under the restriction that it must lose only 6 per cent. velocity in performing that work.

We will assume that our fly-wheel may be 10 feet in diameter taken on the line of mean velocity in the rim. The corresponding circumference is 31.416 feet, and hence 31.416 × $\frac{72}{60}$ will be the mean velocity on this circumference. It is 37.7 feet per second.

We find the mean pressure of the area *m h g e* with sufficient accuracy by measuring the dotted ordinates in the usual manner of averaging pressures in working up steam indicator diagrams. So determined we find it to be 5,395.87 foot-pounds.

The mean velocity of the rim being 37.7, a variation of velocity amounting to 6 per cent. of this makes the maximum velocity = 38.831 feet per second, and the minimum velocity = 36.569. Using the same formula as was used in the practical example given in the previous article, we have

$\frac{64 \times 5{,}395.87}{(38.831)^2 - (36.569)^2} = 2{,}025.4$ pounds,

which is the weight of rim required.

When the problem is complicated by the use of several cylinders applying their power at different crank angles it becomes necessary to consider the rotatory effort exerted at the different angles; while this does not render the method inapplicable it makes it more tedious. The discussion of this phase is reserved for the future.

It is often desirable to know whether a fly-wheel on hand will answer for use in a machine to be constructed. Having decided what percentage of variation is proper to be allowed, it may be that the fly-wheel will control the fluctuation to something less than this percentage; or it may be that it will permit the velocity to vary only so little more than is desirable that it will be practicable to use it.

Having determined the weight of the rim, the maximum velocity and the work, the performance of which will reduce the velocity the allowed percentage, which latter is to be computed in feet per second, then from an equation given in the previous article, $\frac{64 \text{ times the work in foot-pounds}}{\text{Weight of rim}}$ = square of velocity, we may derive the following rule:

Rule.—From the square of the maximum velocity subtract the work in foot-pounds multiplied by 64 and divided by the weight of the rim, and extract the square root of the remainder. The result will be the minimum velocity. The difference between the maximum and minimum velocities will be the variation of the velocity of the wheel.

Example.—In designing a machine it is found that the work tending to reduce the velocity of a fly-wheel making 120 turns per minute is 1,100 foot pounds. In the yard is standing a fly-wheel 4 feet in diameter measured on the line of mean velocity in the rim, the rim weighing 800 pounds. A variation of 4 per cent. in the velocity is allowable. We are to determine whether the wheel in question will answer the purpose.

The maximum velocity of the wheel, if it be used, will be $\left(12.5664 \times \frac{120}{60}\right) + 0.02$ of the product, and it is 25.636 feet per second.

Also $\frac{64 \times 1{,}100}{800} = 88$; hence

$\sqrt{(25.636)^2 - 88}$ = minimum velocity = 23.858 feet per second.

25.636 — 23.858 = 1.778.

The mean velocity is $12.5664 \times \frac{120}{60} =$ 25.1338 feet per second, and the fraction $\frac{1.778}{25.1338}$ will give the percentage of velocity which will be absorbed by the wheel if it be used. This is found to be a little over seven per cent. The wheel is therefore inadmissible.

Class in Steam Engineering.

The evening class in steam enginering of the Young Men's Institute at 222-224 Bowery, New York City, will open on the evening of Wednesday, October 5th, 1892, and will continue throughout the winter until May, 1893. The object of this class is the instruction of all employed or expecting to be employed in the practical operation of engines, boilers and machinery, and desirous of gaining a knowledge of or improving themselves in the theoretical branches, rules and practice of steam engine, boiler and power transmission calculations. Special attention is given to all information not attainable in shop practice, and to such matters required by examining boards, and particularly essential to the engineer, fireman, machinist, boiler maker and the like, in the proper performance of his daily duties. It is the only class of its kind in the city. The course at present extends over two winters. The class has been in operation four winters, and is a recognized success. Mr. William H. Weightman will be in charge as usual.

Chief Smith, of the Department of Transportation, has secured the promise of two monster locomotives to adorn the entrance to the Railway World's Fair passenger station, inside Jackson Park. The Rogers Locomotive Works has agreed to furnish one, and the Brooks Company the other. Each locomotive will weigh 160,000 pounds, and be mounted on a pedestal. They will stand one on each side of the main entrance.

It is generally believed that there will be another advance in the price of anthracite coal in October.

AMERICAN MACHINIST

Communications relating to the reading columns should be addressed to the Editor. All other communications should be addressed:

AMERICAN MACHINIST,
203 BROADWAY, NEW YORK.

PUBLISHED WEEKLY AT
203 Broadway, New York.

HORACE B. MILLER, Pres't and Bus. Manager.
LYCURGUS B. MOORE, Treas. and Sec'y.

F. F. HEMENWAY, Editor and Mech. Engineer.
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$4.00 a year to Other Countries, postage prepaid.

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Transient, 35c. per line, each insertion.
"Business Specials," 50c. a line.

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Publishers' Agents, New York.
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Subscriptions received in Paris, France, by E. TERQUEM, 31 Boulevard Hausmann, at 20 francs per annum, postpaid.

Subscriptions received in Australia by W. WILLIAMS, 17 and 21 Fink's Buildings, Melbourne.

NEW YORK, SEPTEMBER 15, 1892.

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The Iron Hall Failure.

We hope that few of our readers are involved in the collapse of the Order of the Iron Hall, but fear that its popularity among shop men in certain sections of the country has drawn some of them in. If so, then all we seem able to do at present is to express our sincere sympathy for them, with the hope that they will in the future so far follow our often-repeated advice as to keep entirely clear of all concerns which promise to give returns for investments very much above what the best business men can secure except under exceptional circumstances.

The immediate cause of the failure of the Iron Hall was gross mismanagement and dishonesty upon the part of some of its officials, but it is probable that this only hastened its failure, and that the time would have come when it would have been unable to meet its obligations, even had it been well managed and its funds handled in a straightforward manner.

There is almost any amount of capital which is constantly seeking investment at a rate of interest far below the returns promised members of the Iron Hall. But the holders of this capital require a reasonable assurance that their money will be forthcoming upon demand or at the expiration of the time agreed upon. If by ordinary commercial operations any such returns as those promised by this and similar orders could be earned, this surplus capital would be immediately invested and the rate of interest would rise.

But there will be those who will say that the Iron Hall has actually carried out its agreement in many instances; which is true enough, but by no means demonstrates that the principle upon which it did, or was supposed to do business was a sound one, nor that all who go into such schemes will get their money. In fact, where some men get such returns as have been given by such means it proves that others will get nothing.

We have refrained from alluding to this particular case so long as our criticisms could possibly tend towards causing a loss by any innocent member, but we have repeatedly warned our readers against concerns that promise something for nothing, or inordinate returns without proportionate risks.

It is to be hoped that this failure may help to bring about the day when workingmen cannot be induced to invest their money except in enterprises that are confined within the lines of legitimate business operations.

Cultivation of the Thinking Habit.

Mr. Allen's articles on the Cultivation of the Inventive Faculty continue to attract interest, and the number of solutions received, and the care evidently taken in the preparation of most of them give conclusive testimony to the fact that he and the AMERICAN MACHINIST are doing much to stimulate thought and the habit of thinking among mechanics. And when a mechanic is induced to form the habit of thinking there is no telling where he will stop, nor what eminence he may not attain.

Some of those who send in solutions have been more or less trained in the line of work involved, but many others are working machinists and pattern makers with little or no experience in designing or devising mechanical structures. And there is encouragement for these in the fact that their efforts, generally speaking, compare very favorably with any others. In fact, the best solution of a recent problem is by a quite young machinist who is almost entirely self-taught.

A correspondent—a resident of a foreign country—suggests that the interest in these problems might be increased by offering prizes of some mechanical nature for the best solutions, and gives some very strong reasons for the adoption of the plan. Upon the whole, however, we think there are stronger reasons for not adopting it, and that very few who are not attracted by the problems for their own sake will be induced to study them by the hope of a direct reward other than that which always comes to him who exerts his mental faculties in the solution of any serious problem.

We shall hope that many young men will date the beginning of their rise in the profession from the publication of these articles designed to promote thinking; and if so then we shall be sufficiently rewarded, and we believe that all who sincerely study the problems, and then carefully compare the different solutions as presented, endeavoring to pick out the best points in all of them, will be benefited far more than they possibly could be by any prize which we could give. We hope to see the interest in this matter still further increase.

George W. Richardson.

George W. Richardson died at his home in Bridgeport, Conn., at 8 P. M., Thursday the 1st inst. He was one of the best known mechanics in the country, but above this was his reputation as a man. Richardson was born at Poultney, Vt., in 1828. As years count he was comparatively young at the time of his death, but measured by actual work accomplished he lived longer than most men live. Richardson's father was a cabinet maker in the days when that was one of the best trades in the country. Young Richardson, when a lad, learned to use such tools as the cabinet makers used. The knowledge thus gained was of great benefit to him in after years. He was an expert wood turner and pattern maker.

Working in wood did not suit Richardson. The fascination of working in metal was so strong with him that when a mere youth he went to learn the trade of a machinist, I believe at Newton Upper Falls. After serving his time he worked in different shops, his skill as a machinist always insuring him a "job." In those early days in the trade, reputation spread, and he was well known throughout New England as an expert machinist.

Railroad mechanical matters had a charm for him, and he gravitated to Salem, N. Y., the mechanical headquarters of a road which is now a branch of the Delaware & Hudson Canal Company's system. From there he went to Troy, N. Y., as foreman for the Troy & Boston. This, I think, was in 1858.

Previous to this—I think when he was learning his trade, or a little later—he joined a class in mechanical drawing, taught by the late J. C. Hoadley. A few years since, Mr. Hoadley referred, in a communication to the AMERICAN MACHINIST, to the outcome of the members of this class. He referred in high terms of praise to Richardson, and so plainly as to perfectly identify him, although no names were mentioned in his article.

Richardson's mechanical skill was of great service to the Troy & Boston road. He brought out systems of making repairs, and changed things to correspond; out of disorder he organized order. Increasingly hard service and high steam pressure pulled the driving wheels loose, making no end of trouble. Richardson evolved the plan of boring the wheels a taper; it was a very slight taper, I think about $\frac{1}{64}$" in the length of the hub. What I do remember about it is that after the weight of the axle, vertically suspended, came on the fit we used to draw them in four and one-half inches. I never knew such a fit come loose. He also devised a series of hand reamers for bolts, pins and the like. These reamers were tapered $\frac{1}{32}$" in seven inches and made so that one would follow another, making it entirely practicable to follow up one with the other, until a perfect hole was obtained.

Somewhere about this time Richardson's health failed, and by the advice of his doctor—and a rather strong inclination in that direction—he went running on the road.

Richardson soon conceived an intense dislike for the ordinary safety valve, loaded with a "Sutlers" balance, and he soon conceived a valve that would raise from its seat upon the intimation of blowing off, permitting the free escape of steam. I helped him to make and apply the first of these valves; we did the work at night. This valve, first patented in 1866, has in an improved form since become famous.

Later, Richardson invented what he termed a balanced slide valve for locomotive and other engines. This valve is very favorably known to-day. In its construction correct principles were employed, and stated with great clearness in the patent papers, the preparation of which Richardson personally superintended.

I am not writing of Richardson as if he were a perfect man. There are no such men. He was quite positive in his views, just as I imagine most men are positive if they know anything. But there was this about him, when in his positiveness he thought he had wronged a man he took the earliest opportunity to rectify the matter. His ways of doing this were his own ways, but there was never room for doubt as to what he meant. George W. Richardson never willfully wronged any one without making restitution.

I could fill all the pages of an issue of the AMERICAN MACHINIST with anecdotes and circumstances to this end. He was a man, and having said this there is little more to say.

He leaves a wife to whom he was married while young, and a married daughter. Richardson will always be known from his invention of the "Pop" safety valve, which is so well known as to leave nothing to be said of it.

F. F. H.

Literary Notes.

VALVE GEARS FOR STEAM ENGINES. By Cecil H. Peabody, Associate Professor of Steam Engineering at the Massachusetts Institute of Technology.

The aim of the author is to give engineering students instruction in the theory and practice of designing valve gears for steam engines, and give the learner a firm grasp of the principles, and some facility in their application. The whole subject is divided into six chapters, treating on the plain slide valve, shifting eccentrics, link motions, radial valve gears, double valve gears, and drop cut-off valve gears. In the presentation of the elementary principles, geometrical and analytical methods are used; the latter method necessarily involves equations which may be perplexing to the ordinary reader, but with a knowledge of algebra and trigonometry there will be no difficulty in understanding them. In the application of these principles, geometrical methods are used exclusively, so as to conform with the usual practice of laying out valve gears by construction, making the whole subject a drawing board process, as it should be, and this, in connection with the many fine illustrations, will enable the reader who is willing to make a serious effort to acquire a fair knowledge of laying out valve gears. The illustrations are on folding sheets, which, when spread out, expose to view the whole illustration, thereby avoiding the annoyance of turning over leaves to find letters of reference, and interrupting the close attention and study which complicated subjects, such as valve gears, demand. Common and well-known processes have been used, and the author has rather avoided than sought novelty. Some things, however, he believes are presented for the first time; for example, the combination of a skeleton model with the construction for laying out link motions and other irregular or complicated gears, and several examples of double-valve gears; the latter are introduced mainly to show the scope of the methods used. The book is published by John Wiley & Sons, 53 East Tenth street, New York. Price, $2.50.

DYNAMOMETERS AND THE MEASUREMENT OF POWER: A Treatise on the Construction and Application of Dynamometers; with a description of the Methods and Apparatus employed in measuring Water Power. By John J. Flather, Ph. B., M. M. E., Professor of Mechanical Engineering, Purdue University; Author of American Edition Wilson's "Steam Boilers."

In this little book of 215 pages, including a copious index, the author treats the important subject of the measurement of power by means of dynamometers in an exceedingly comprehensive and interesting manner, and the value of the work is further enhanced by a concise description of the construction and principles of action of the various types of dynamometers employed, which, with a very few exceptions, are now in use. The

whole subject is treated in the following order: Determination of driving power. Friction brakes. Absorption dynamometers. Transmission dynamometers. Power required to drive lathes. Measurement of water power.

Considerable of this subject has appeared in a series of articles recently published in the AMERICAN MACHINIST. The convenient form in which these and much additional matter are now published adapts the whole for the use of technical students, engineers, draftsmen, practical men and all others interested in the accurate knowledge of the amount of power absorbed by different machines, and in estimating the cost of power for running various plants with economy and efficiency. It is not only an excellent book of reference, but it is also one which can be studied with profit by those who are desirous of obtaining a knowledge of the construction of the various types of dynamometers and their practical application. It is published by John Wiley & Sons, 53 East Tenth street, New York. Price, $2.

THE JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS for August contains very interesting and instructive articles contributed by prominent engineers. The subjects treated on are: An investigation of the mode of action of ship-formed solids on the water in which they move. The chase of the "Itata." Method of molding in loam and casting the cylinders for the cruisers "Cincinnati" and "Raleigh," at the Navy Yard, N. Y. Feed-water heating. Comparison of the propellers of some United States Navy ships. Formula for proportioning riveted joints and calculating the width and thickness of butt straps for steel boilers. Trial of the paddle steamer "Ville de Douvres." Experiments with basic steel. The whole forms a valuable contribution to technical literature, and should be in the hands of all engineers.

QUESTIONS AND ANSWERS.

Questions of general interest relating to subjects discussed in our columns will receive attention in this department. The writer's name and address should always accompany the question. Neither correct initials nor location will be published when there is a request to that effect. If questions are enclosed with a business communication, they should be written on a separate sheet.

(384) V. M. C., Saginaw, Mich., writes: Please answer through your columns whether water containing 246 grains of salt and other matter, but principally salt, per U. S. gallon, would be injurious to a boiler. The water has a brackish taste, but does not create a thirst after drinking. *A.*—It will be decidedly injurious to the boiler, and the injurious effect may not only be aggravated by the kind of salt, but also by the nature of the other matter.

(385) J. C., Bridgeport, Conn., asks: At what speed should a condensing engine with a high pressure cylinder 4 inches diameter, low-pressure cylinder 7 inches diameter, stroke 6 inches, and with a steam pressure of 150 pounds, drive a boat 40 feet long, 9 feet beam and 4 feet draught of good model, with a wheel 3 feet diameter and 6 feet pitch? Steam pipe is 1¼ inch, and exhaust pipe 2 inches diameter. *A.*—About 8 miles per hour, provided the steaming capacity of the boiler is sufficient. We should advise you to reduce the pitch of the propeller.

(386) E. J. B., Philadelphia, Pa., writes: Please inform me in what issue of your paper a description is given of the method of treating castings to make them smooth for the use as patterns. *A.*—See answer to Question 289, in our issue of July 14, current volume. 2. I have been told by a party that a cam can be so arranged about a crank-pin of an engine that there will be no dead center, and that they had seen it in use on a steamboat engine. The party could not describe it. Kindly inform me if you have seen or heard of it. *A.*—We have never seen nor heard of it.

(387) H. R., Newark, N. J., writes: I am under the impression that the English patent law fees have been reduced recently. Kindly inform me through your columns when this reduction was made, and how much. *A.*—The new law goes into effect October 1 of this year. The changes are only in the taxes, not in the first cost of patent. The total of taxes under the old law amounted to £150, payable after the end of the fourth year. Under the new law the total amount is £95, beginning with £5 before the commencement of the fifth year, and increasing £1 per year until the patent runs out, making a saving of £55 in all.

(388) E. E. R., ——, Pa., writes: Please inform me through your columns what the size of steam ports should be for an engine with cylinder 3 inches diameter and 6 inches stroke. *A.*—Steam ports 1¼x¼ inch; exhaust port, 1¼x½ inch. 2. What should be the weight of fly-wheel? *A.*—About 50 pounds. 3. Where should steam be cut off? *A.*—About half stroke. 4. I wish to connect this engine to boiler placed 80 feet from it. In this case should I use a larger steam pipe than when the engine is placed close to the boiler? *A.*—Yes. 5. What size of steam and exhaust pipes should I use? *A.*—Steam pipe, ¼ inch diameter; exhaust pipe, 1 inch diameter.

(389) C. W. M., ——, asks: Will you tell me of some solution for washing the grease from nipples which have been cut with lard oil. We have been using a hot solution of sal-soda, but it is not entirely satisfactory. *A.*—The solution of hot sal soda is quite generally used for this purpose, and is usually satisfactory where it is in a kettle provided with means for keeping it hot by steam, and the articles are placed in a perforated vessel, which allows of their being soused about in the solution. You do not specify in what particular the work is unsatisfactory, which, of course, lessens the chance of our being able to help you. Perhaps some of our readers have used something better than sal-soda, and will tell us of it.

(390) J. H. J., Harvey, Ill., writes: Please give me a rule or formula by which I can ascertain the size and diameter of packing rings for steam cylinders, also for air cylinders. *A.*—We presume that you refer to piston packing rings. There is no rule for determining these sizes; experience is the best guide. In our answer to Question 382, September 8, current volume, we have given the size of cross-section for such rings. In regard to the diameter, we have been accustomed to turn the cast iron rings, before they are cut, about one-quarter of an inch larger in diameter than that of the cylinder, and then cut enough out of the ring to spring them to the diameter of the cylinder. This answers well for cylinders up to 20 inches diameter. For larger ones, the cast-iron piston packings should be turned somewhat proportionately larger in diameter. A. E. Seaton in his Manual of Marine Engineering recommends an excess of 1 per cent. of the diameter of cylinder.

(391) E. S., ——, Pa., writes: I would like to have your opinion, as well as that of your readers, on the following subject: Suppose a steam boiler has 7 holes ¾ inch in diameter punched in the shell near one head; the holes are located in a circle whose diameter coincides with that of the boiler, and have about 2¼ inch pitch. Now how much do these holes weaken the boiler? We will suppose that the holes have been closed by rivets. *A.*—These holes will weaken the sheet, but they will not weaken the boiler as a whole, because the strength of the boiler will depend on the strength of the riveted joints, which is less than that of the solid sheet. 2. Will these holes weaken the boiler more than two ¾ inch holes 3 inch pitch for brass rivets? *A.*—No. 3. Will the same number of holes, same pitch with a 2⅞ hole in the center, but reinforced with a plate $\frac{5}{16}$ inch thick and 2 inches wide all around the hole, be a source of weakness, and to what extent? *A.*—The 2⅞ inch hole, like the rivet holes, will weaken the sheet, but if the reinforcing plate is thicker than the boiler sheet and the width of the plate all around the hole is at least 3 times the diameter of the rivet, the boiler as a whole will not be weakened.

(392) W. A. R., Chester, Eng., writes: I wish to harden a cast-steel roller 9" diameter by 22" long. The surface of this roller is covered with small projections of a pyramidal shape—¼" square at the base and $\frac{5}{16}$" high. I wish the temper to be very even over the surface, and to be of medium grade. Can you give advice as to how to do this safely? *A.*—Such a job cannot be safely done in a forge or in any open fire. It should be heated in a muffle, and it is better also that this muffle should be part of a gas or other furnace if practicable. If you have not or cannot provide such facilities, it would be better to send the job to an establishment to be hardened where they have facilities for such work. In quenching, we should use pure rain water, and plunge the roll vertically into it to a depth of at least three feet, holding it there until the outer surface is sufficiently cool, after which it may be removed and submitted to a gentle heat until the temperature becomes uniform throughout. In making such a roll as this, it should always be considered whether or not it is practicable to make it in short sections in the form of rings to go on an arbor. Sometimes there are reasons why this cannot be done, but where it is possible to do it, it diminishes the risk of loss in hardening very much.

(393) C. A., Toronto, Canada, writes: You have been kind in settling difficulties, and I thought you would help me out in solving a problem. I have just purchased a book on practical physics, by Steward and Gee, called "Elementary Practical Physics." The trouble is this: There are words or abbreviations in the theory of the balance, such as cos, sin., tang., sec. Please explain what these words mean. *A.*—Cos. means cosine of an angle; sin., the sine of an angle; tang., the tangent, and sec., the secant of an angle. These are all trigonometrical functions, and you will find them explained in any elementary trigonometry. 2. Do you think this book is the proper one for a young machinist to study who wishes to learn designing and engineering? *A.*—The book is undoubtedly a good one, but it is too difficult for you to commence with; there are much simpler ones. In our issue of February 19, 1891, page 4, we have indicated a course of study for young machinists, and have also given the titles of suitable books. This course we believe you can easily follow, provided you make an earnest effort to do so. Of course, it takes considerable time and labor to go through this course, but you will certainly not regret it. 3. I have gone through practical geometry and projections. How much would the articles on practical drawing, published in your journal, help me? *A.*—They will help you to make working drawings—a very important branch in engineering. 4. What is the cost of these papers? *A.*—$4.65, or single copies 5 cents each, postpaid.

(394) R. T., ——, Pa., writes: I would like to inquire for the proper size of steam and exhaust ports, size of cam and travel of slide valve for an engine with cylinders 30 by 36 inches. *A.*—The size of steam ports depends on the piston speed, which you have not given. You can find the area of these ports by the following rule. Multiply the square of the diameter of the cylinder in inches by the piston speed in feet per minute and divide the product by 7,639; the quotient will be area of each steam port in square inches. If, for instance, the piston speed of your engine is to be 500 feet per minute, then the steam port area will have to be $\frac{20^2 \times 500}{7,639} = \frac{400 \times 500}{7,639} = 26.18$ square inches. The length of ports is arbitrary, for large engines it is generally from ¾ to ⅞ of the diameter of the cylinder. If now it is decided to make the ports 17.5 inches long then the width of ports will have to be $\frac{26.18}{17.5} = 1\ 49$, say 1½ inch. The travel of the valve will depend on the lap of the valve, and is found by multiplying the sum of the lap and the width of the steam port by 2. If, for instance, the lap of the valve is 1¼ inches then the travel of the valve will be $(1\frac{1}{2} + 1\frac{1}{4}) \times 2 = 6\frac{1}{2}$ inches. We presume that by the word "cam" you mean eccentric. The eccentricity of the eccentric will have to be one-half of the travel of the valve, making the throw equal to the travel, provided that there is no rocker with arms of unequal lengths interposed between the eccentric and valve. The width of the exhaust port is found as explained in our issue of May 1, 1890; it is frequently twice as wide as the steam ports.

(395) H. B., Philadelphia, Pa., writes: I have been employed at mechanical drafting for about a year and a half and am desirous of becoming a first-class draftsman, but on account of the lack of practical experience in a machine shop I find that it goes much against my success. As I am dependent on my earnings I cannot take a course in a machine shop, on account of the low wages upon which I would have to begin, and for similar reasons I cannot go through a scientific course at college. What books would you advise me to read, and what course of study to pursue in order that I may gain at least the principles or theoretical knowledge of molding, blacksmithing, pattern making, and such things that contribute to my success? *A.*—In our issue of February 19, 1891, we have indicated a course of study relating to the theoretical branches, which, if followed, will greatly aid you in your advancement. For books treating on the practical branches of the business we should advise you to examine the books in the public library and select such as will suit your purpose. Also follow closely articles from practical men which frequently appear in technical journals. In time you will probably be frequently called into the shop, and this will give you excellent opportunities for observing how work is done. Compare this with what you have read and form your own conclusions of the best way of doing work. If, in addition to this, you form the acquaintance of practical men with whom you can spend a part of your leisure time in a pleasant and profitable way by the interchange of ideas, you will lay up a fund of practical knowledge which will be extremely valuable to you. As you advance you may be able to lay some of your earnings aside, so as to enable you to take a few years' instructions in the shop, and thus acquire that practical experience and knowledge which all draftsmen and engineers should possess.

BUSINESS SPECIALS.

Transient Advertisements 50 cents a line for each insertion under this head. About seven words make a line. Copy should be sent to reach us not later than Saturday morning for the ensuing week's issue.

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"Indicator Practice and Steam Engine Economy." By F. F. Hemenway. Contains plain directions for using the indicator and making all required calculations from the diagram, also the principles of economy in operating steam engines and current practice in testing engines and boilers. Price, $2 postpaid. Published by John Wiley & Sons, 53 East Tenth Street, New York.

Practical Details of Die-sinking—I.

BY B. F. SPALDING.

GETTING READY.

The practice of forging in dies is by no means a novelty. Many years ago, iron-work was thus formed. In the description of a famous wrought-iron hinge, made in the 13th century, Mr. Crane, author of "The Smithy and the Forge," says the smith had top and bottom tools, in which were the impress of the ornaments desired, and that the striker would "bump them up" with a few blows of a 10-pound sledge. It is quite probable that such dies have been used from the remotest times; the use of them is one of nature's suggestions. In 1786 there was an unsuccessful attempt made in France to manufacture interchangeable muskets; die-forging was one of the accessories relied upon to carry out the scheme, but so little were those who practiced it skilled in the business, that it was abandoned "on account of the number of pieces spoiled and cracked by improper swaging." Captain Hall is the author of drop-forging as now practiced; he carried it into successful operation in the government works at Har-

per's Ferry, in the early part of this century.

The practice of drop-forging is increasing every year, and has become quite general, yet there are always some shops where it is new, and the sinking of the impressions and indentions in the blocks of steel, which are to be the matrices in which the plastic metal is to be formed is new in such localities, and information on the subject is welcome. A description of some methods by which this work can be done may bring out more valuable suggestions.

In such shops as have been mentioned, the facilities for doing this class of work are often quite primitive, and it is necessary in them to perform by hand the operations which, in more fortunate establishments, have for the last 35 years been more easily executed by machinery especially designed and made for die-sinking.

There are two principal modes of die-sinking, one of which is by "typing," and the other by cutting. By the first and probably the original method, a piece similar in form to that which it is desired to produce is driven into the block which is to be the die—the block being softened by heat for this purpose. By the second method, the stock of the block is cut away until the exact shape of the impression has been scooped out. The first method requires the greatest artistic and technical skill to insure such perfect results as it is now necessary to obtain, and for this reason the second method —that of removing the stock by cutting it away—is the one most in vogue.

Although no machine can exceed in beauty and accuracy the work which can be done by that trained and experienced craft of manipulation which we call "skill," it is a fact that the mechanic who knows just what the work he has to do will look like—or ought to look like—when it leaves his hands, can produce such work quicker by machinery, and with less experience, better than he can without machinery. I have seen a man produce dies quickly and nicely with a common edging machine, who would work long and earnestly at the vise to make dies which he never was proud of.

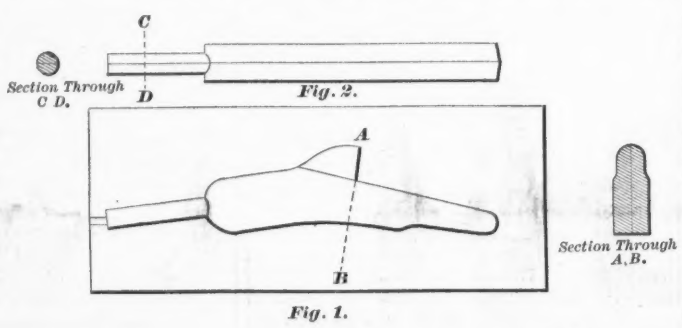

Fig. 1. *Fig. 2.*

Let us take a pair of dies, for an example, for making the piece shown in Figs. 1 and 2, and follow the processes necessary to make them in a shop where only ordinary tools are available. The figures represent (1) the plan view of the piece lying on a block ; (2) the side view, with sections as indicated.

The first thing to be considered is the fitting of the dies to the drop, or other machine, where they are to be used, in order to be certain that the points which the impressions in each die are laid out and set off from may coincide when they are set in the die seats where they are to be used, so that the prints in the upper and lower dies may correspond.

It is advisable, when convenient, to have the faces of the dies correspond in size all around. There is some advantage in having them of the same length and the same width, although measurements should be made from one side and one end, rather than from both sides and both ends, or from either promiscuously.

The faces should be made smooth enough to wipe, and then the clean and bright surfaces may be washed over with a saturated solution of sulphate of copper (bluestone in water). If the faces are clean when the solution is applied, they will be coated with copper, and can readily be marked upon with lines which will show distinctly.

Put the dies together as they are to go, and dot with a prick punch the left side and front end of each die just as they stand. Put two dots in each place on the lower die, and one dot in each place on the upper die. With very little more trouble, letters may be used instead of this conventionalism.

Take the dies apart, and lay out a center line lengthwise and one crosswise on the face of the lower die. Lay out lines on your upper die to correspond with these, measuring from the dotted sides of your dies.

You may work from either side or either end, but it is a good plan to have a certain rule of your own to follow in this respect in making dies; it prevents confusion, and if you stick to it you never have to puzzle yourself about which side you laid out this or that pair of dies from. You will know that you laid out this pair just as you always laid them out. The double dotting or lettering enables you always to distinguish the upper from the lower die.

Now you want a pattern, which is the model of the piece for which the die is to be sunk, not a model of the finished piece, but of the forging that is to come out of these dies. It will frequently pay to make such a model, even if you have the finished piece to go by, for it may save errors of calculation which are liable to creep into allowances which are simply figured out. Discrepancies in a model may be detected, which in figures would escape observation.

In making the model, first settle upon what amount of "draught" is to be allowed for the metal to draw out of the die. If the sides of the impression should be made perpendicular to the face, the work would stick in the die. About 5° from perpendicular is generally enough to allow. After the die has been worked for a while, the edges will batter or draw over, so that the work will stick, and then the edge of the impression is set back by the repairer ; the tendency, therefore, is always to gain in draught as the dies wear, so it is not best to allow too much draught at the start; sometimes even less than 5° is admissible, but if the work does not come out freely with 5° it may be 7°, for it is a serious injury to dies to allow hot work to lie in them forced up against their edges, as they then absorb heat rapidly, and cooling quickly afterwards, induces checks to form in them. Make the model to cover the draught.

Having the model made, the next thing to provide is the profile template, which is a piece of thin sheet metal. If it is an "outside" template, it is cut to the exact shape of a section of the model, on the line of its division, the part of the model on one side of the dividing line being to go into the upper die, and the part on the other side into the lower die.

The dividing line, in many cases, will be the line on which molders would part their molds, the upper die representing the "cope," and the lower die the "drag." Inexperienced die sinkers often have great difficulty in conceiving just how their die must look when it is done. Not unfrequently the designer of the die also can catch a good suggestion from the same proceeding. It pays to get as clear an idea as you can of what you are about to make before you commence to make it. Drawings are easier made than molds, but one mold shows more than three drawings. It is easier to make changes in a sand or plaster mold than it is in a steel one.

After an "outside" template is finished to the proper shape, it is laid on the lower die, and adjusted to the position where it seems desirable to locate the impression, and the center lines of the die are faintly drawn upon it as it lies; these lines are then transferred to the other side of the template, which is then laid on the upper die, guided by the corresponding lines there. It will then be seen whether the impression—if laid out on this plan—is as well placed in the upper die as it is in the lower one, or if a better arrangement can be made. If the present disposal is satisfactory, the lines across the profile are drawn permanently; it is laid on the lower die, by the lines, and its shape accurately marked off on the die. Before removing it, it is well to dot it lightly with two prick punch marks, or the equivalent letter, on the front end, as a precaution against getting it on the upper die in the same, instead of in a reverse position sidewise, and to insure that it is *not* reversed endwise.

A die cannot be sunk too nicely, but a workman may be too nice in sinking a die. The precautions herein mentioned are not too great—they will save time and mistakes.

Some die sinkers prefer to use an "inside template" instead of an "outside" ; it may be more work to make it, but in some respects it is more convenient. A good inside template may be made from a piece of sheet steel—No. 17 or thinner—the thinner the better if it is stayable. The sheet should be somewhat larger than the face of the die. Mark and cut it out in the proper position, and drill small pin holes just to the guiding edges of the dies; insert pins in these holes, and when you lay out your dies place the template on the die, shoving it up so that the pins will touch the guiding edges. This gives but a small chance for errors in laying out the impression.

After the outlines of the profile are correctly drawn on the coppered surface of the die, go over them with a fine prick punch, and lightly and very carefully dot them.

The operations thus far described are generally to be performed, whether the die is to be sunk by hand or by machinery; unless there are dies enough to be made to pay for originating a mother die or "former," in which the guide-pin of the copying machine moves and a cutter moving simultaneously reproduces the generating form. Even then the work on the preliminaries is rather more than less than has been told.

Thirty years ago a die sinker was given this problem to solve, viz.: to devise a machine which would do such work as to sink a pair of U. S. musket hammer dies direct from the male former; in other words, to make a copying machine, the guide-pin to which should bear upon and be led by the model hammer itself, in sinking both the upper and lower dies.

A few years ago he informed me that the problem was solved.

MANUFACTURES

The oil-mill at Terrel, Tex., is nearing completion.

Work has commenced on a flouring-mill at Claud, Texas.

The erection of a flouring-mill at Queen City, Tex., is being discussed.

A new shirt factory has been added to the industries of Glassboro, N. J.

The Common Sense Engine Co. has removed its entire plant from Springfield, O., to Muncie, Ind.

The Columbia Power Company, near Farmingdale, N. J., will soon manufacture smokeless powder.

The McNeil & Tinder Electric Company, at Winchester, Ky., will need equipment for a general machine shop.

Messrs. Leslie and Lorin Packard are to build a steam mill near their old water mill, at Highland, in Warren, Me.

The Laconia Car Company, of Laconia, N. H., will increase its plant sufficiently to double its manufacturing capacity.

The Genesee Fruit Company, of Bouckville, N. Y., is to enlarge the capacity of their barrel factory from 150 to 300 barrels per day.

Coates & Co.'s tin-plate mills, at Baltimore, Md., which were destroyed by fire several months ago, have been rebuilt, and will resume operations in a few days.

The new boilers of the Fort Worth (Texas) Compress Company have arrived, and are being put into position, and the company is prepared to do a big business this season.

A new silk factory is to be established at Oswego, N. Y., that will employ several hundred hands in the manufacture of both ribbons and dress goods, by Tietz & Strassberg.

The Lehigh Company is building a 1,000-foot extension to its West Superior (Wis.) coal dock, which will give it a capacity of over 400,000 tons.

While boring for an artesian well at Mount Arlington, Morris County, N. J., workmen struck a deposit of graphite, or black lead, which is so hard that it dulls and chips the drills.

The Gage mill, on Fountain street, Pawtucket, R. I., will soon be put in operation by the American Yarn Manufacturing Company. Many improvements have been made in and about the building.

Owing to a very general improvement in the iron market, the steel works at Stowe, near Pottstown, Pa., will increase its force of operatives several hundred men, and run on double time, beginning August 24.

The Boonton (N. J.) Iron and Steel Company intends to enlarge its plant in the near future. A bar and merchant-mill will be erected where the old plate-mill stood. The repairs which the mill has been undergoing have been finished.

W. Roodberg, of Worcester, Mass., who has been engaged in manufacturing razors for a number of years, is in Rockford, Ill., for the purpose of organizing a company to operate a razor factory. It is the intention to organize a company with a capital stock of $50,000.

The Webster & Perks Tool Co., of Springfield, O., are quite busy on bolt cutters, some of which they are sending to Europe. Their reducing wheel for indicating steam engines, which we illustrated some time since, has also met with a very favorable reception.

A stock company is being formed at Los Angeles, Cal., for the manufacture of a new metallic vehicle wheel hub, the invention of Mr. Wells H. White, of Los Angeles. The intention is to manufacture in one or more eastern cities, and to prepare for an extensive display at the World's Fair.

Geo. F. Archer, Jr., at his mill at Parks Pond, Clifton, Me., has sawn this season a considerable amount of long lumber, and about 500,000 shingles. He will soon saw out the frame for a house which he is to build this fall for his own occupancy. During the winter he will get out the timbers for a new mill, which he proposes building next season.

The latest enterprise with which Isaac C. Atkinson, the president of the Atkinson House Furnishing Co., is associated, is the purchasing of the water power of the East Branch Falls, at Conway, N. H. Mr. Atkinson is a regular summer guest there, and he has in contemplation an electric plant to furnish light to North Conway, Intervale, Lower Bartlett and Jackson.

The main building of the new shops of the Shepard Bridge Works Company, of Havana, N. Y., is 210x60 feet. It will have a wing of ample proportions at right angles. The interior will be supplied with derricks, tram cars and other paraphernalia for handling iron, arranged with a view to convenience and expediency. It is understood that 100 hands will be employed in the new plant.

A Vermont firm—Skinner, French & Co.—have purchased of J. Manchester Haynes, of Augusta, Me., the township in Franklin County, next south of Lowelltown on the C. P. Railroad, No. 2, Range 5, and will build a large steam saw-mill. The mill will be about a mile from the Caswell station in Lowelltown. The boarding house and some other buildings are nearly completed, and the railroad company will put in a siding as soon as possible. The mill will be ready for operation next spring.

E. B. Estes & Sons, who have extensive wood-turning mills in Maine, New Hampshire and Vermont, are rebuilding their Farmington factory, which was destroyed by fire a week ago, and are putting up a building considerably larger than the one destroyed, and the same will contain fifty of their patent automatic lathes. Every lathe going into the new building will be an improvement on the lathes destroyed by the fire, and the new mill will have double the capacity of the old. This firm is the most extensive wood-turning house in this country.

The Elmira Electric Illuminating Co. are building a new plant at Elmira, N. Y., and have placed the contract for the iron roofs with the Berlin Iron Bridge Co., of East Berlin, Conn. The dynamo room will be 66 feet wide by 177 feet long, covered with the Berlin Company's patent anti-condensation corrugated iron roofing, the floor space being controlled by a ten-ton traveling crane. The boiler room will adjoin the dynamo room and will be 32 feet wide by 75 feet long. When completed this will be one of the finest electric light plants in New York State.

The business of manufacturing novelties and special machinery, which was formerly carried on at 60 and 62 Murray street, New York, by Mr. John M. Lambert, will in future be conducted by the John M. Lambert Manufacturing Co. Mr. Lambert will remain in charge of the affairs of the new company. The business of manufacturing electrical appliances and experimental work which has been carried on by J. Jones & Sons at 39 Vesey street, New York, will be combined with that of the John M. Lambert Mfg. Co., though the office and store of J. Jones & Son will continue at 39 Vesey street.

Wheeler, Campbell & Co., of Glasgow, Scotland, are starting a lace factory at Columbia, Pa., under the title of the Columbia Lace Co. Two brick buildings, each two stories high, and a boiler house are being put up, giving altogether about 50,000 square feet of floor space. The Scotchmen have brought over their lace-making machinery because they say such machinery is not yet made in this country, but the bleaching machinery and the boil-

ers and engines are of American make. The present buildings have room for 12 sets of machinery, and the mill is expected to employ about 300 persons.

A factory at Sherman, Tex., manufacturing seamless bags, has proved a great success. Six bales of cotton are used per day, and the output is 3,000 finished bags. The most improved machinery has been used in equipping the factory, and includes a complete electric light plant, and a ventilating system arranged so as to drive cool air throughout the buildings in warm weather, and warm air in the winter. Another establishment in the same town is making cotton seed-oil. The company controlling this are now erecting a larger mill, which will require over 400 tons of cotton-seed per day.

Machinists' Supplies and Iron.

NEW YORK, September 10, 1892.

Iron—American Pig—We quote Standard Northern brands, No. 1 Foundry, $15 to $15.50; No. 2, $14 to $14.50; Gray Forge, $13 to $14. Southern brands of good quality are obtainable at $14.25 to $15 for No. 1 Foundry; $13.50 to $14 for No. 2; and $12.75 to $13.50 for Gray Forge.

Scotch Pig—Coltness is quoted at $21; and $20.25 for Eglington.

Antimony—The market is dull and lower. We quote L. X., 11¼c. to 11½c.; Cookson's, 12c. to 12¼c.; Hallett's, 10¼c. to 10½c.

Copper—The market is quiet, and quotations are nominal. Lake Copper can be bought at 11¼c. in round lots. Casting Copper is held at 10⅝c. to 11c.

Lead—There is no important change in the market. Some business is reported at 4.17½c. for car lots, but the general quotations are 4.15c. to 4.17½c.

Lard Oil—Prime City is quoted at 62c. to 63c.; Western on spot at 61c.

Spelter—The market is quiet. Holders are quoting 4.70c. in a nominal way.

Tin—The market is steady, with no change in quotations of 20 25c.

WANTED

"Situation and Help" Advertisements only inserted under this head. Rate 30 cents a line for each insertion. Seven words make a line, and the address as desired to appear should be included. Copy should be sent to reach us not later than Saturday morning for the ensuing week's issue. Answers addressed to our care will be forwarded.

Wanted—Agents in every shop for machinists' fine tools, sliding calipers, etc. E.G.Smith,Columbia, Pa.

Wanted—Position as foreman boiler maker; reference furnished. Foreman, AM. MACHINIST.

Mech draftsman, good designer, wants situation. Address BOX 71, AMERICAN MACHINIST.

Machinist and draftsman, 15 years' experience, wants pos. as foreman. A. B. C., AM. MACH.

Mechanical draftsman desires position; references. Address BOX 72, AMERICAN MACHINIST.

First-class draftsman, having practical experience on land and sea, desires position; not constantly at board pref'd. Reliable, AM. MACHINIST

Wanted—Position by designer of hydraulic machinery and tools; 8 years draftsman, 8 machinist and tool maker. Box 68, AMERICAN MACHINIST.

Wanted—Foreman for brass finishing dep't of brass works for steam & water goods; only men of exp. need apply. Box 80, AMERICAN MACHINIST.

Wanted—Exp. foreman for med. size mach. shop and foundry doing gen. jobbing; state age, experience, wages and references. Maryland, AM. MACH.

Wanted—Three planer hands, 3 lathe hands, and 2 floor hands, all first-class men. Sioux City Engine Works, Sioux City, Iowa.

Wanted—Two good machinists and two molders for dry and green sand molding. Apply or address to Kenney & Co., Scottdale, Pa.

Wanted—Position by young man having experience as machinist and tool maker; 7 years' a foreman on aut. mach. & mach. tools. Box 83, AM. MACH.

Wanted—Five good non-union machinists at once to take the places of strikers; strike not on account wages. Jenney Electric Motor Co., Indianapolis, Ind.

Position wanted as foreman or machinist; 14 years successful exp. in Germany and this country. H. Froehlich, 444 Norwood Ave., Cleveland, O.

A practical experienced erecting engineer desires position as superintendent on the erection of heavy machinery or new plants, mill-wright work, etc. Box 81, AMERICAN MACHINIST.

Wanted—First-class mechanical draftsmen. Address Box 82, AMERICAN MACHINIST.

A pattern maker wishes a position; some southern State preferred; will accept any good offer; is a good workman; steady, sober and reliable. Address C. H., care AMERICAN MACHINIST.

Wanted—By pattern maker, exp. and reliable, steady pos. and good class of work; would take pos. in tech. school as instructor in pattern making and fine wood-work. Robert Rockwell, Lynchburg, Va.

A practical mechanical engineer, well up in engine, boilers, machinery, and possessor of my own steam indicator, is open for engagement; best of references. Box 78, AMERICAN MACHINIST.

Wanted—At once, 6 mech. draftsmen of exp.; no others need apply. Apply, stating age, experience, and wages expected, to Western Avenue office of Thomson-Houston Electric Co., Lynn, Mass.

Young man of 23 with three years' experience in machine shop work—bevel and spur gear cutting especially—and three years as manager small mfg. business, including experimenting, designing, drafting, etc., wants to make a change from present employment to small machine shop as draftsman, and where business experience can be utilized if required. Box 79, AMERICAN MACHINIST.

Wanted—Foreman for our steel plate blower depart.; one who has had experience at boiler or heavy sheet iron work; must be thoroughly competent to handle gang of 50 men at work to best advantage; state fully all your experience and length of time you have occupied the different positions you offer as references; also what salary you would expect. Address Buffalo Forge Co., Buffalo, N. Y.

Bradley Hammers.

If you plate, draw, square, taper, swage, collar, spindle, or do any manner of die forging in iron or steel, a BRADLEY HAMMER will soon pay for itself, by what it will save over any other similar tool. Send for circulars.

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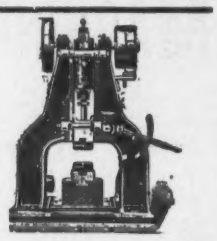

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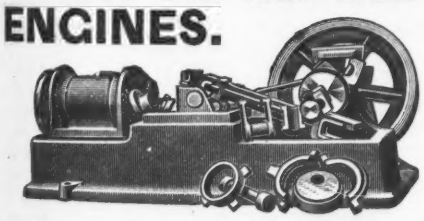

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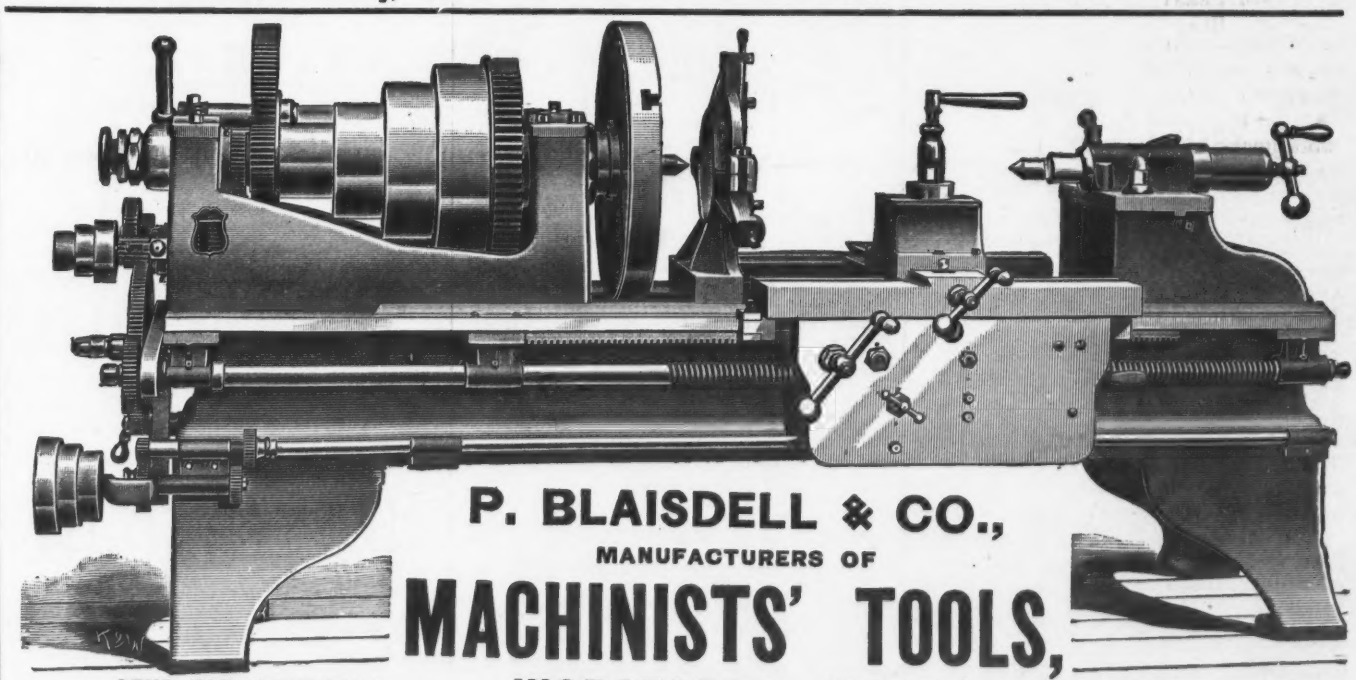

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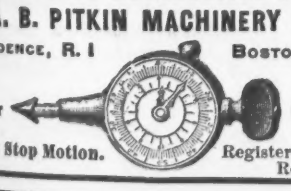

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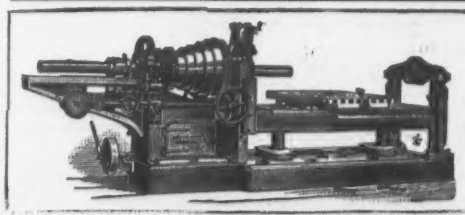

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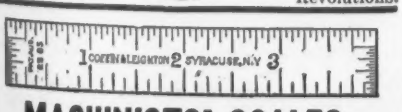

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Wanted—1st-class man, successful in managing men, and thoroughly acquainted with screw machine work, to take superintendence of a machine screw factory in Western city. Address Queen City, AMERICAN MACHINIST.

✢ MISCELLANEOUS WANTS ✢

Advertisements will be inserted under this head at 35 cents per line, each insertion. Copy should be sent to reach us not later than Saturday morning for the ensuing week's issue. Answers addressed to our care will be forwarded.

Cheap 2d-hd lathes & planers. S. M. York, Clev'd, O.

Auto. Steam Flue Cleaners. Kelley Co., Erie, Pa.

Presses, Wire Form'g Mchy. Am. Tool Wks, Clev, O.

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Best Bolt Header in the world for $50. Address C. H. Baush & Sons, Holyoke, Mass.

Engines, special and gen. mach'y designed; ideas developed. A. W. Jacobi, 136 Liberty St., N. Y.

Designs perfected; drawings, patterns and machines made to order. S. Carey, 17 Broadway, N. Y.

Special machinery designed and built. C. F. Langston & Co., 70 N. 4th St., Philadelphia.

Light and fine machinery to order; Foot Lathe Catalogue for stamp. E. O. Chase, Newark, N. J.

Design & superintendence of mach'y. Drawings, etc., made. Claud Mantle, 598 E. 134th St., N. Y.

Screw cutting lathe 9″x54″, with tool and vise, $47; cost, $150. Box 229, AMERICAN MACHINIST.

Wanted—A specialty to manufacture either heavy or light work. We have splendid facilities. Address "Manufacturers," Rockford, Ill.

For Sale—Half interest in a machinery business, established in Boston twenty-two years; this is a chance of a lifetime. Lock Box 5, Roxbury, Mass

For Sale—A well-located shop down town New York, suitable for brass and light machine work. J. Jones & Son, 39 Vesey Street, New York.

Wanted—Hardware specialties to build. Send 25 cents for "Eureka Knife and Shears Sharpener." W. A. Smith & Co., 227 Eddy Street, Providence, R. I.

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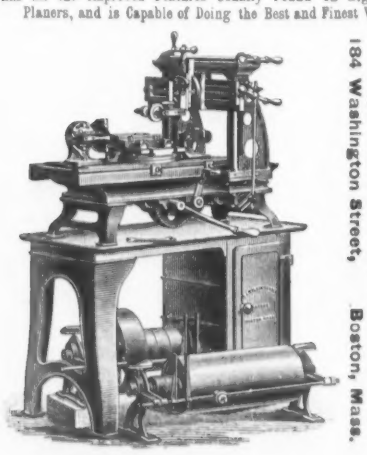

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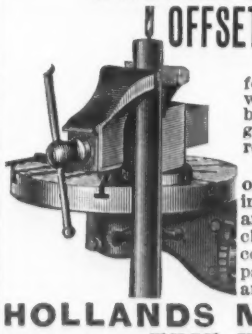

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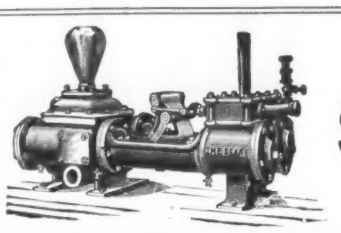

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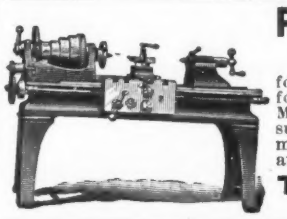

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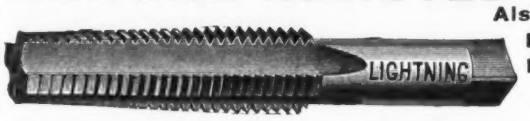

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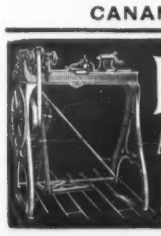

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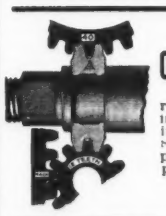

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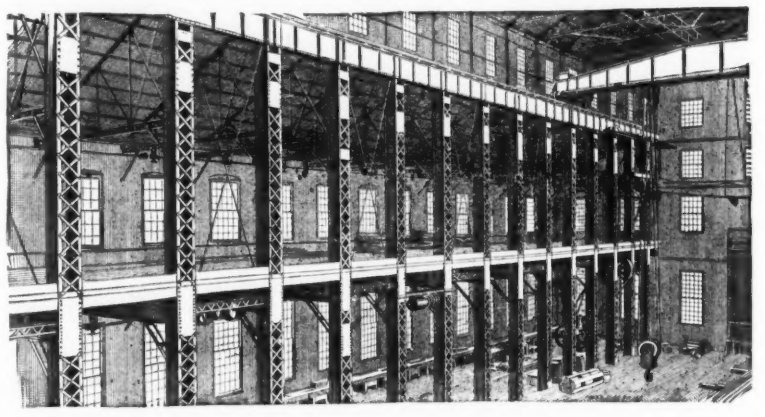

The above illustration is taken direct from a photograph, and shows the construction of an Iron Building designed and built by us for The Newport News Ship Building & Dry Dock Co., at Newport News, Va., and is used by them as a Machine Shop. The photograph was taken looking diagonally across the building, to show the Traveling Crane and the Gallery. The building is 100 ft. in width by 300 ft. in length, the central portion being 50 ft., and the wings each 25 ft. in width. The wings are made two stories high. The central portion is controlled by a traveling crane, and the wings by jib cranes.

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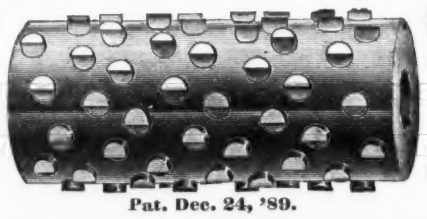

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8	1 1-4	.80
9	1 3-8	.95
10	1 1-2	.95
11	1 3-4	1.10
12	2	1.20
13	2 1-4	1.35
14	2 1-2	1.45
15	3	1.60
16	3 1-2	1.80
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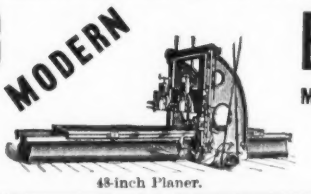

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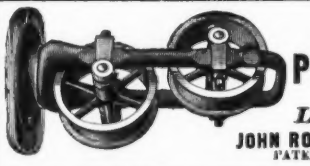

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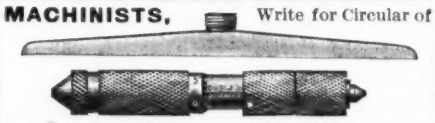

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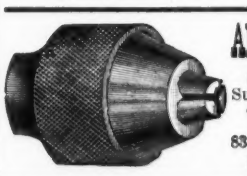

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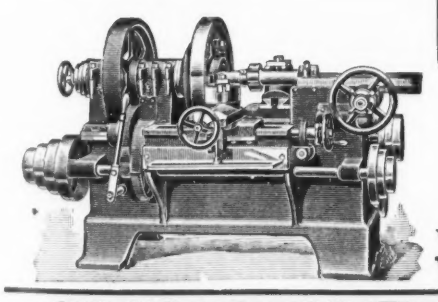

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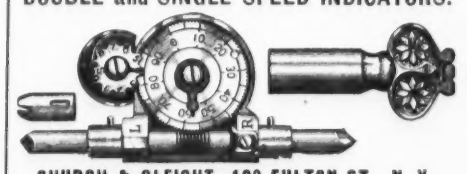

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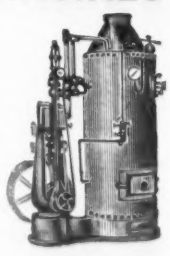

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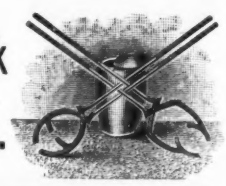

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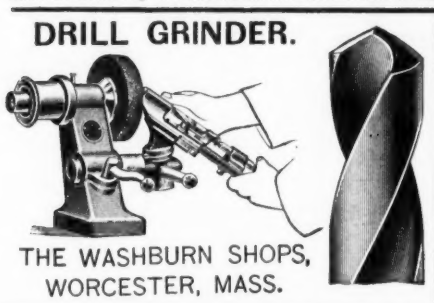

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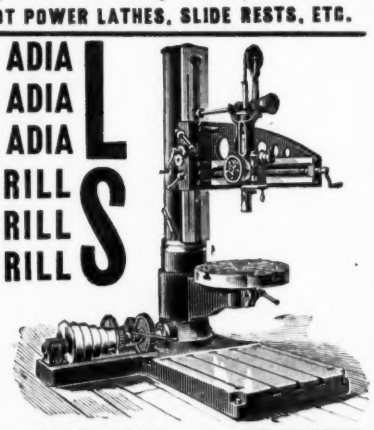

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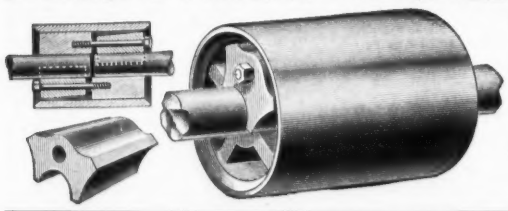

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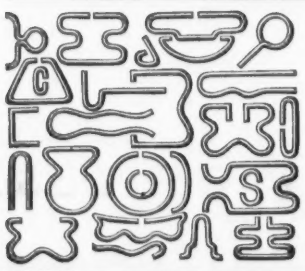

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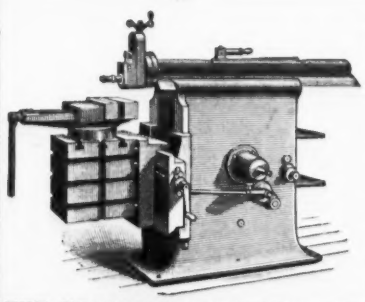

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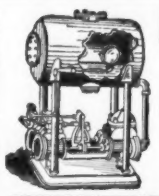

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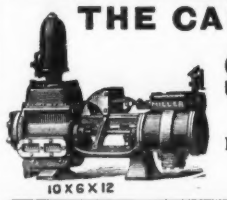
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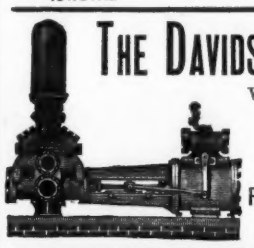

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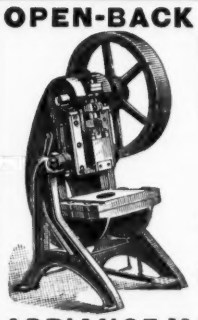

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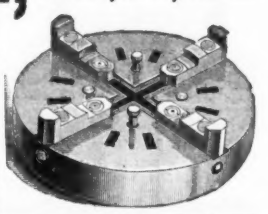
24 "	28 "
30 "	32½ "
36 "	40 "



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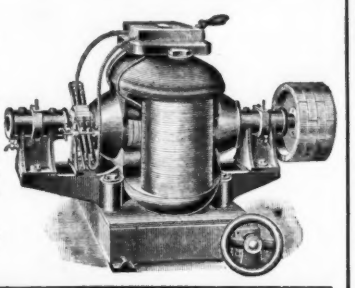

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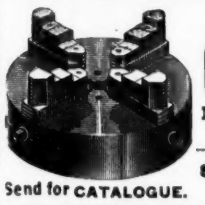

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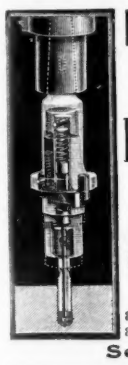

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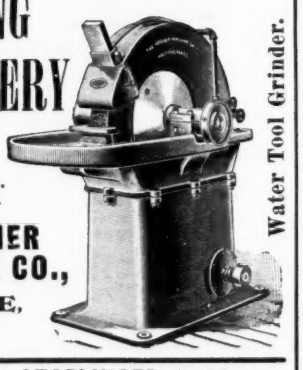

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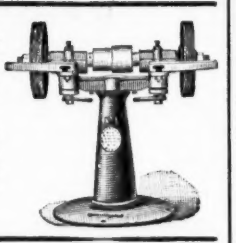

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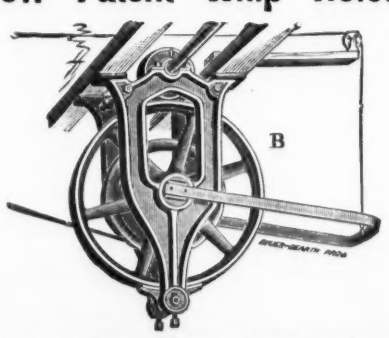

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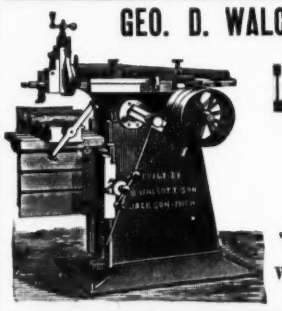

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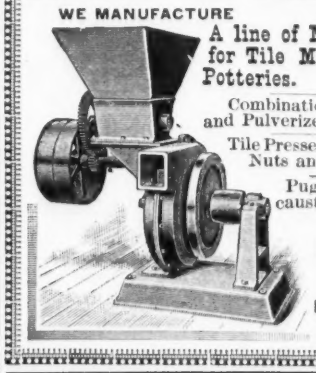

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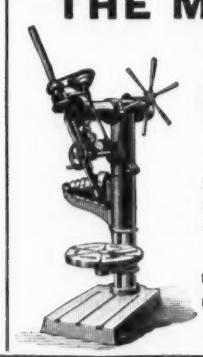

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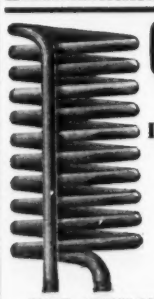

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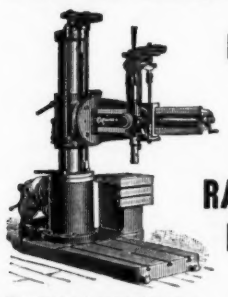

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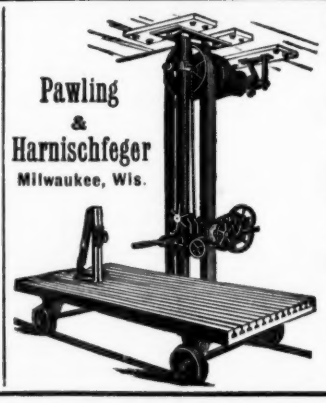

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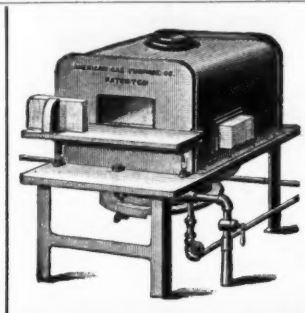

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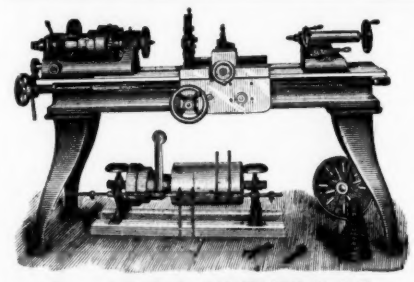

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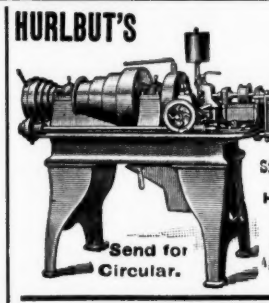

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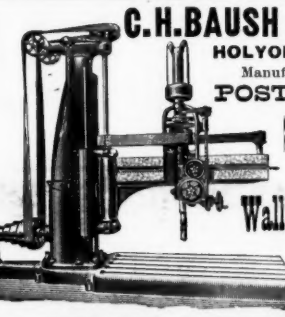

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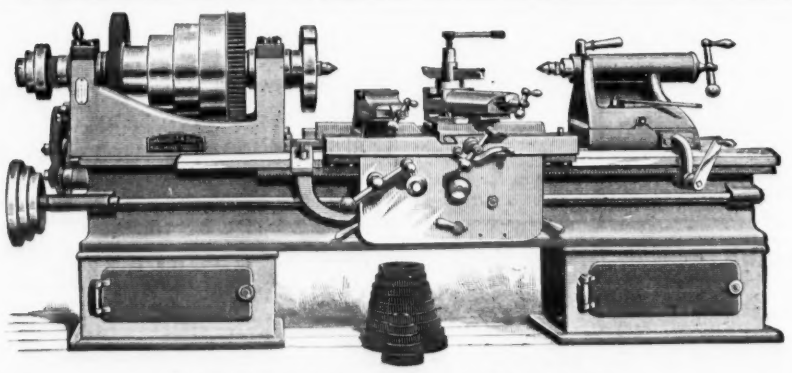

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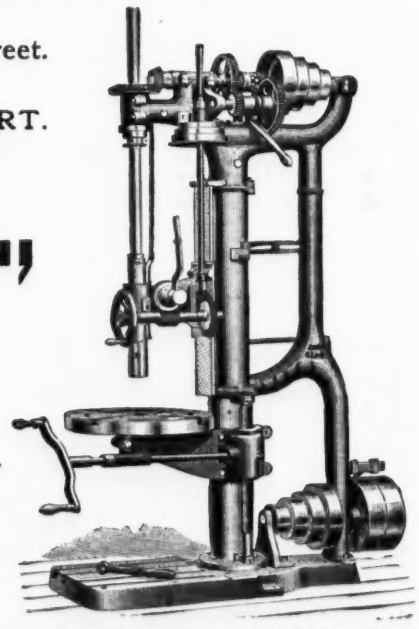

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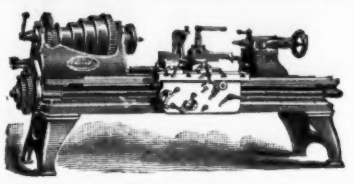

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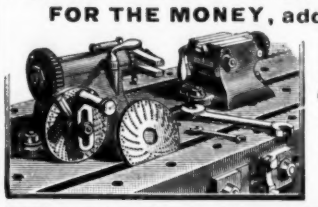

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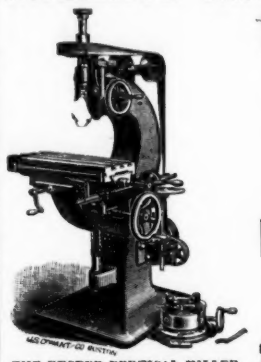

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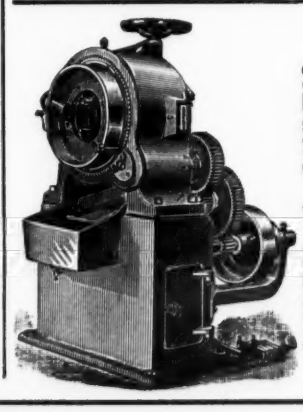

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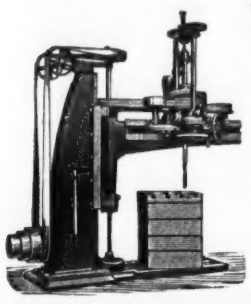

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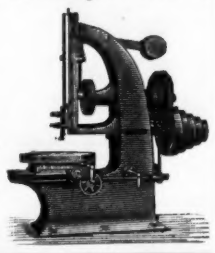

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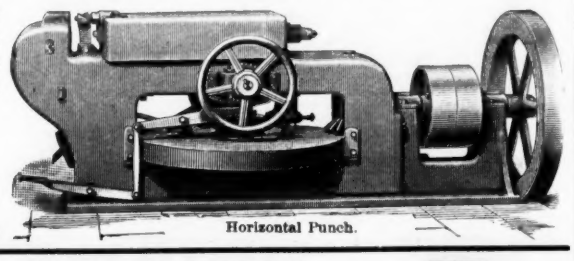
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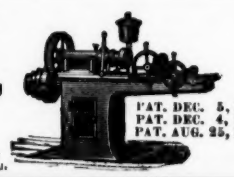
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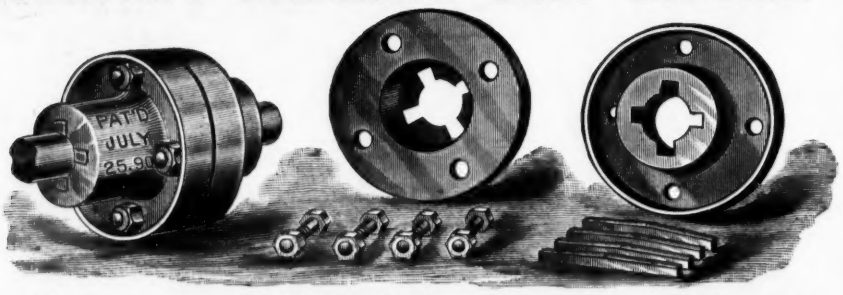

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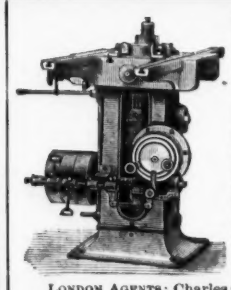

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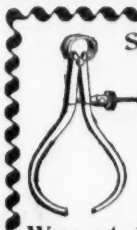

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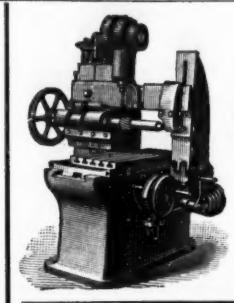

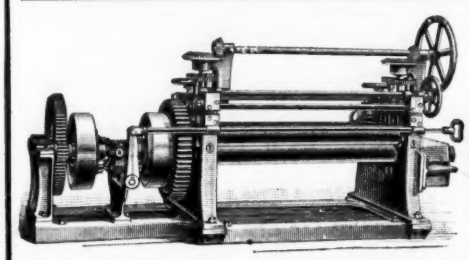

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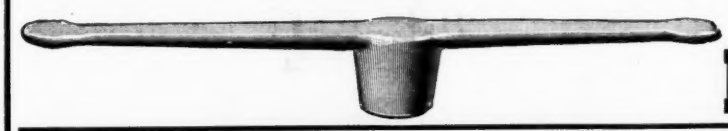

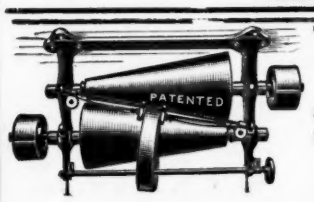

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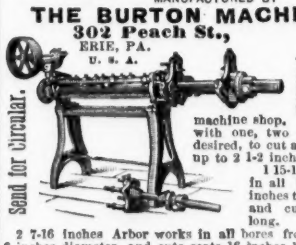
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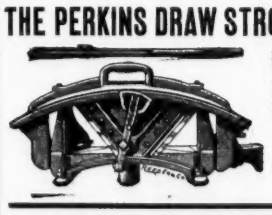

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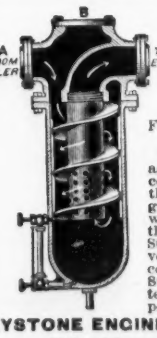

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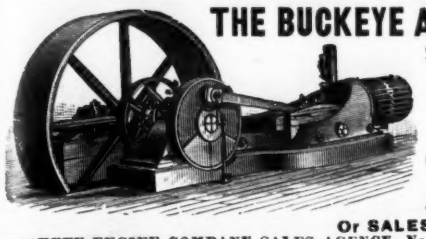

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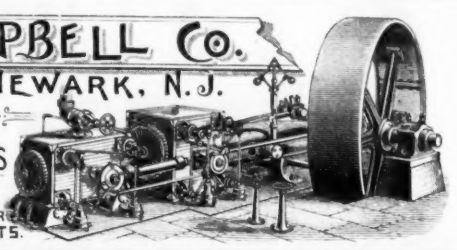

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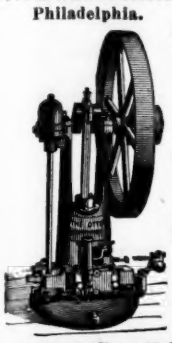

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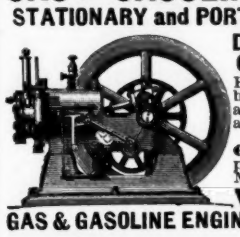

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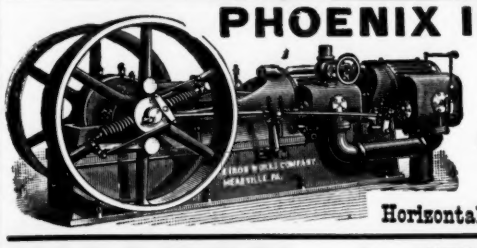

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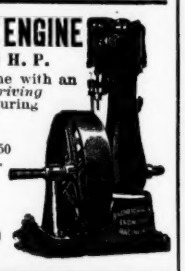

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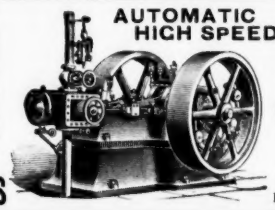

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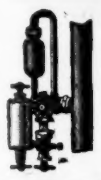

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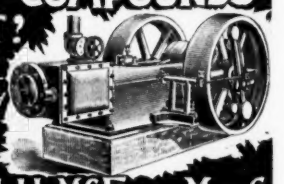

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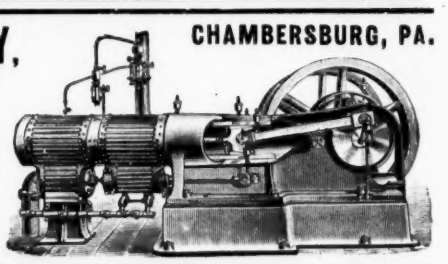

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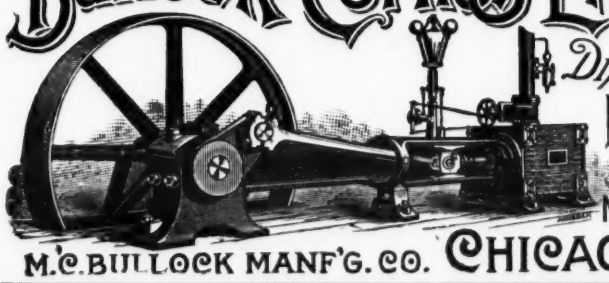

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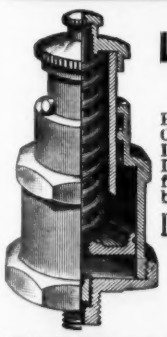

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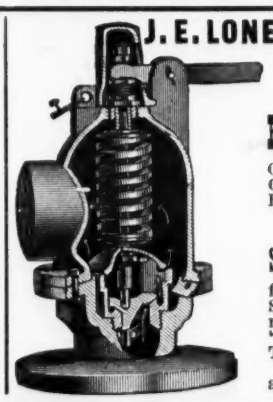

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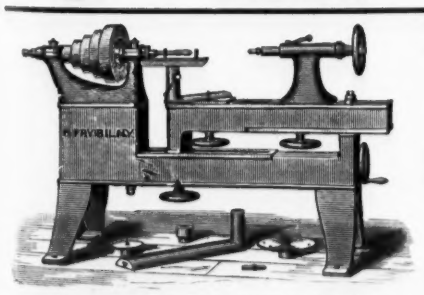

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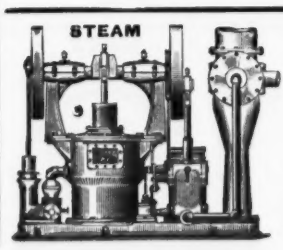

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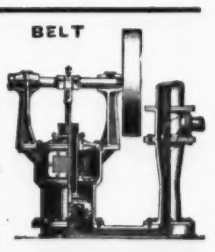

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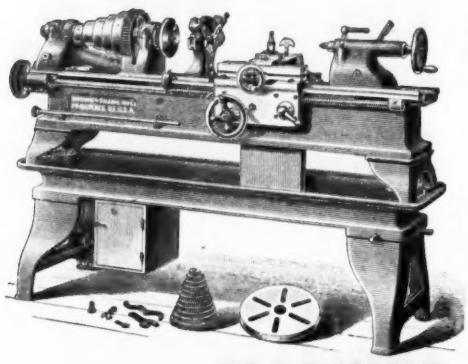

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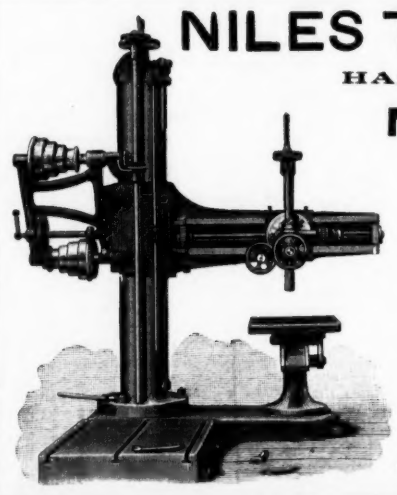

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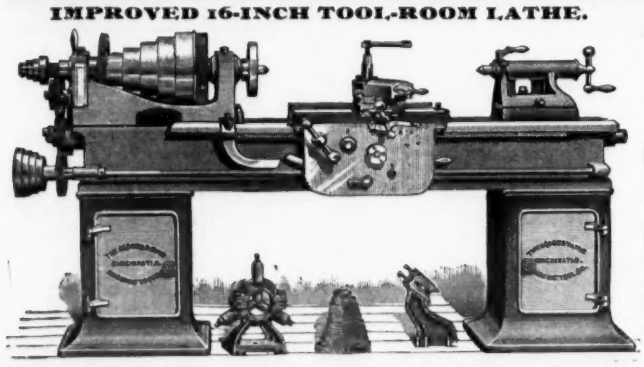

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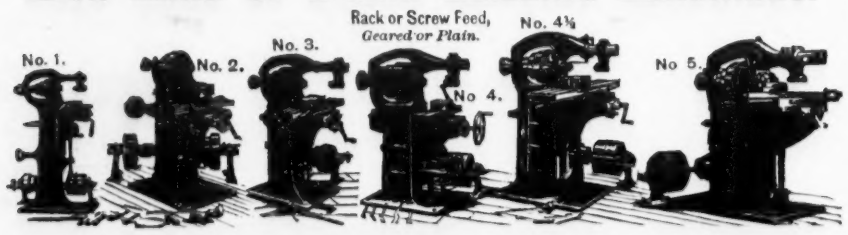

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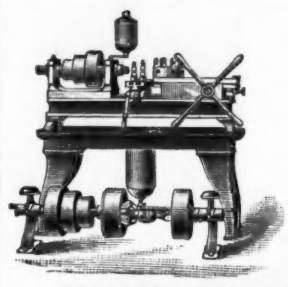

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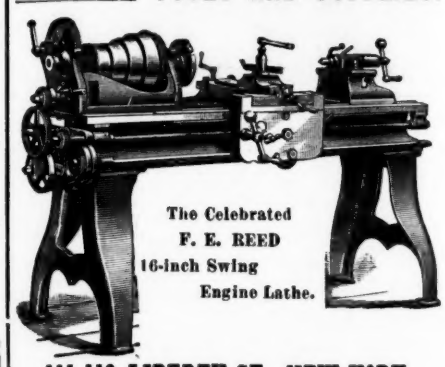

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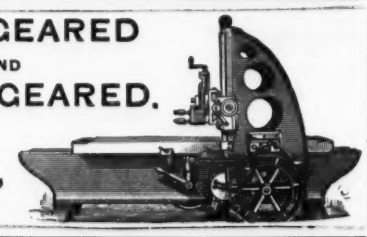

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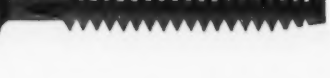

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